GULLEY

Design of an Acyclic

Turbo-Generator Unit

Electrical Engineering

M. S.

1911

DESIGN OF AN ACYCLIC TURBO-GENERATOR UNIT

BY

LAURENCE RICHARD GULLEY

B. S. UNIVERSITY OF ILLINOIS, 1910.

THESIS

Submitted in Partial Fulfillment of the Requirements for the

Degree of

MASTER OF SCIENCE

IN ELECTRICAL ENGINEERING

IN

THE GRADUATE SCHOOL

OF THE

UNIVERSITY OF ILLINOIS

1911

UNIVERSITY OF ILLINOIS

THE GRADUATE SCHOOL

May 31 1911

I HEREBY RECOMMEND THAT THE THESIS PREPARED UNDER MY SUPERVISION BY

Laurence Richard Culley

ENTITLED Design of an Acyclic Turbo-Generator Unit

BE ACCEPTED AS FULFILLING THIS PART OF THE REQUIREMENTS FOR THE

DEGREE OF Master of Science in Electrical Engineering

Ernst Berg
In Charge of Major Work

Ernst Berg
Head of Department

Recommendation concurred in:

Committee
on
Final Examination

DESIGN OF AN ACYCLIC TURBO-GENERATOR UNIT

Contents.

I. INTRODUCTION

The acyclic, or uni-polar dynamo, as it is sometimes called, has long been recognized as an ideal direct current machine, since it generates direct current without commutation by the motion of conductors in a constant magnetic field. The electromotive force in each conductor is then constant and numerically equal to the number of lines of magnetic flux cut per second, divided by 100,000,000. Connections are made through collector rings and brushes and thus any voltage from that of a single conductor to that of all conductors in series may be developed.

This construction affords a very desirable system of voltage control or conversion which should work out most successfully. The possibilities of the type thus seemed unusually good, but practical considerations had still to be taken into account, and it was this phase of the problem which apparently brought failure to all earlier forms of acyclic machines.

The principal difficulties encountered were those of design and also of large current collection at high speeds. These dynamos were usually designed for low voltage, using the rotating disc-shaped member as a conductor, and thus obtaining an E. M. F. corresponding in later machines to the voltage per bar or conductor. A sketch of this type of generator is shown in Figure 1, page 2.

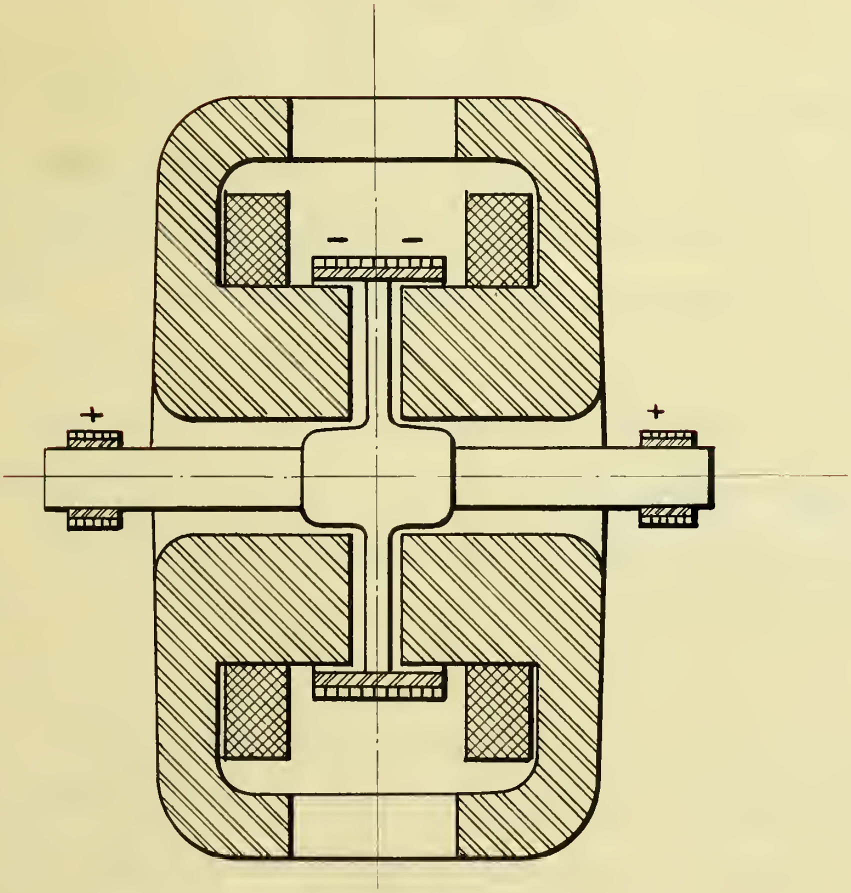


FIG. 1.

Many arrangements have been devised to obtain the desired relation between the moving conductors and the field flux, but all types may be included in two general classes, according to the arrangement of the rotating element as

1. - The radial type.
2. - The axial type.

It will be noted that this classification is not strict in the sense that there are only two types, but rather to classify the main characteristic arrangement of the rotating element. There are in fact, many classes and almost an infinite number of detail arrangements, but all must necessarily be in accordance with the fundamental theory of acyclic machines and will thus be classed in one of the main divisions indicated according to the type of rotating element used. A few examples of various types of acyclic machines are shown on pages 4 and 5.

The characteristic peculiarity of acyclic machines is the annular magnetic field of constant intensity and it is also this feature which has given rise to various incorrect terms such as homo-polar and uni-polar, which indicate that the machine has but one pole - a magnetic impossibility. All electromagnetic machines have at least two poles, and the acyclic type is no exception to the rule, although the poles may not be as readily apparent as in the usual type of dynamo. By reference to the sectional drawings, it will be seen that there are two annular pole pieces, arranged either axially or radially, and that the conductors are arranged to cut the

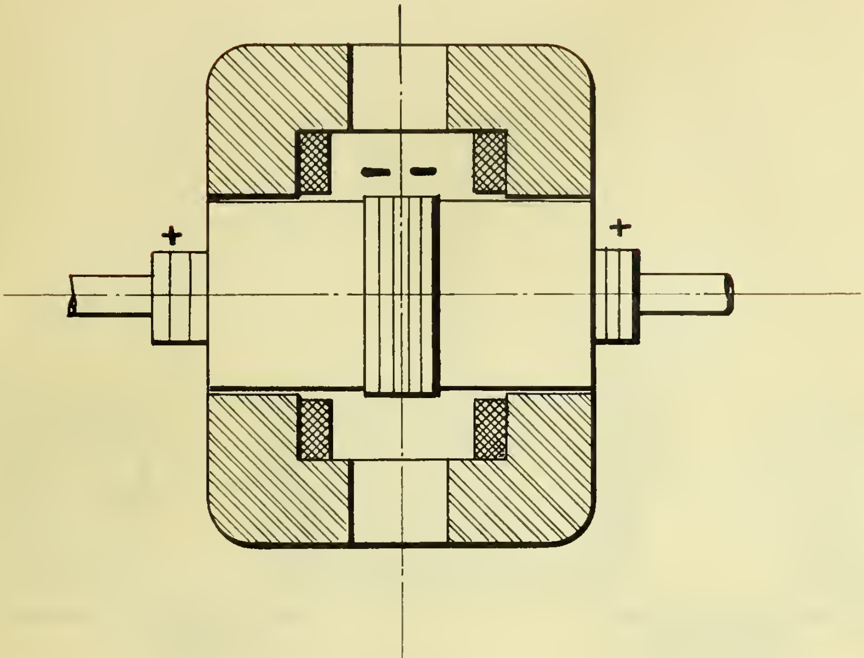


FIG. 2

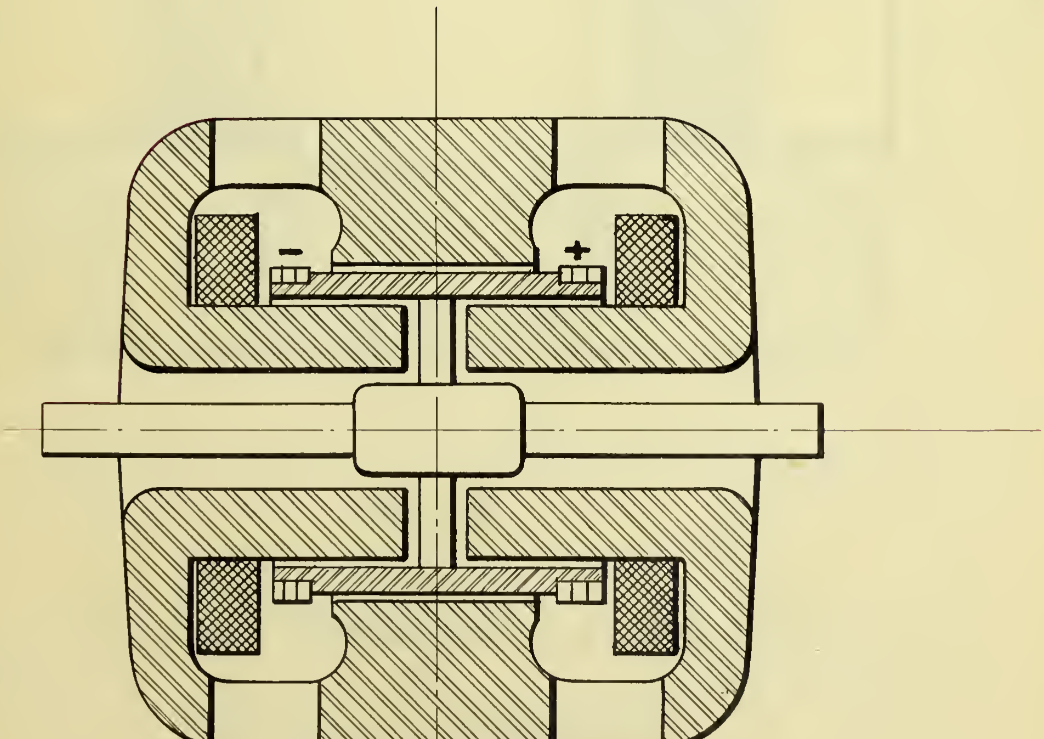


FIG. 3

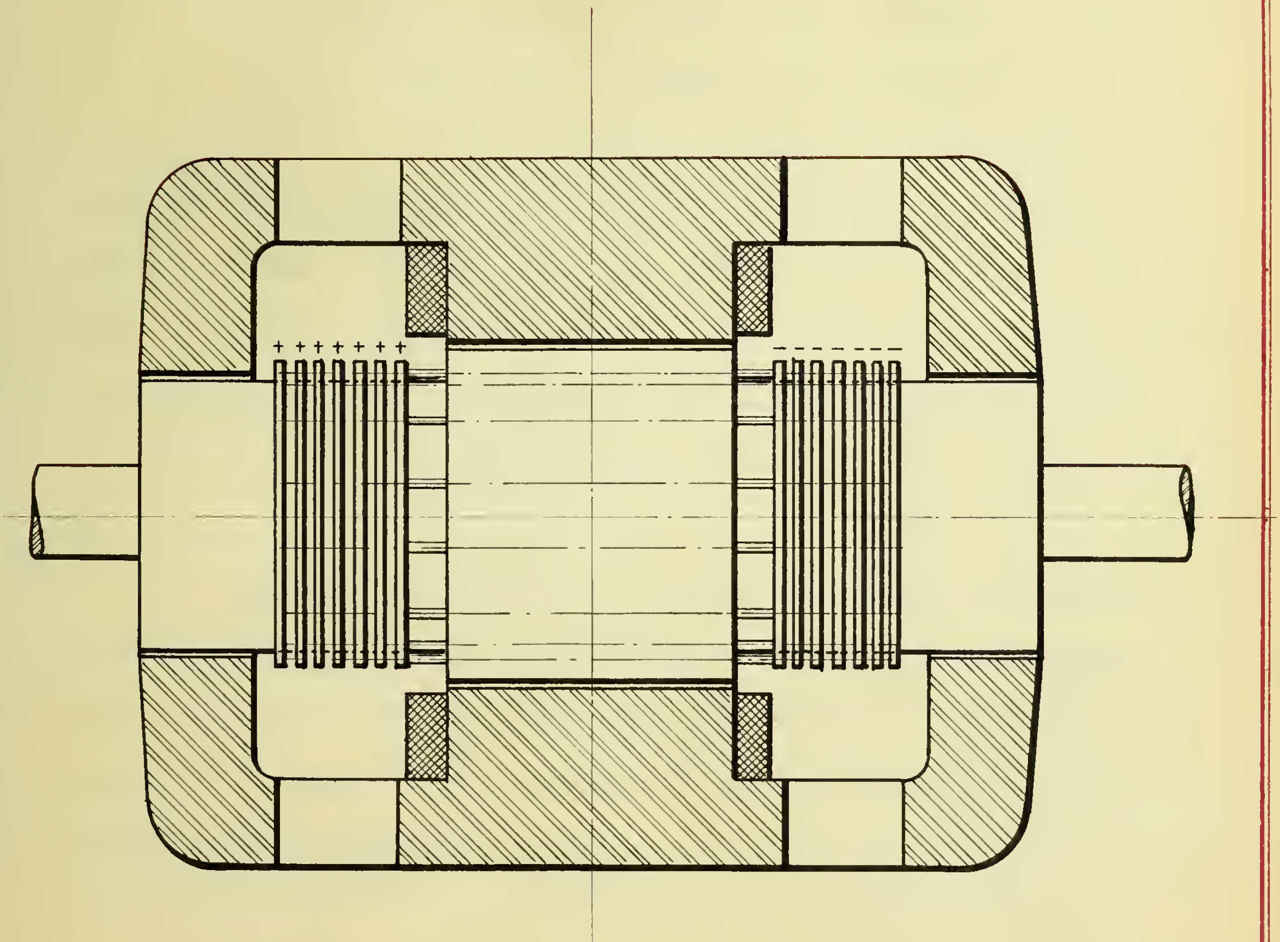


FIG. 4

MULTIPLE CONDUCTOR - AXIAL TYPE
OF
ACYCLIC GENERATOR

uniform field, thus producing a constant voltage with constand speed of rotating element.

In designing an acyclic generator, the question of arrangement and type to be employed will usually be determined by the operating conditions of the machine as well as by the type of prime mover or other driving device to be used. In the present instance, the prime mover is a steam turbine operating at very high rotative speed and thus restricting the permissible diameter of acyclic rotor to a safe value of peripheral speed. This speed condition alone would thus bar the radial type of acyclic generator and necessitate the use of the small-diameter axial type. Figure 4.

Other conditions of voltage, output and operating conditions also govern the design and will be considered later, but the example noted above indicates that the type cannot be chosen at random but is often predetermined by operating conditions.

II GENERAL METHOD OF ACYCLIC GENERATOR DESIGN

The acyclic or "uni-polar" machine, as pointed out in the introduction is, in reality, a two pole acyclic dyanmo of special construction and so arranged that the conductors pass through or "cut" a constant magnetic field maintained

between a stationary annular pole piece and another of opposite polarity which rotates within the first and carries the conductors near its periphery. As the voltage per conductor is comparatively low, the machines are usually arranged with all conductors in series, this connection being obtained by using collector rings and brushes at each end of the rotor. A diagram of this construction is shown in Figure 4, page 5.

The general theory of operation of the dynamo is very simple, as it depends upon the fundamental principle of electricity that a conductor which cuts a hundred million lines of flux per second has induced in it an electromotive force of one volt. In short, the design is based upon the fact that each conductor cutting a hundred megalines of flux per second will furnish one volt of E. M. F.

It is thus apparent that with any given conditions of speed, voltage and number of conductors on the rotor, the machine is practically determined as to size of parts to carry the necessary flux, although we have no idea of its output, which is usually one of the first things desired. This peculiar method of design in which we assume the various operating conditions and determine the output indirectly by calculation, seems peculiar to the design of acyclic machines and is quite contrary to general methods of machine design, which almost invariably originate with the output or duty of the machine and terminate with the size of parts and operating characteristics.

The peculiarity of acyclic design is thus in the uncertainty of the capacity or output of the machine until suit-

able calculations are made. Another peculiarity, especially from the manufacturer's standpoint, is that all machines for any given conditions of voltage and speed will of necessity have the same maximum output and there would thus be only one class and rating of machine for any given operating conditions, as indicated, making the manufacturing problem more simplified than with the common forms of direct current machines.

The calculations which lead to an approximate determination of the output may be outlined in a general method with corresponding equations as follows:

Starting with

$$1 \text{ volt} = 10^8 \text{ lines cut per second,}$$

we have given the desired terminal voltage of the machine, and assuming that each conductor generates approximately 50 volts, we immediately have the number of conductors.

In order to have this conductor generate 50 volts, it must cut 50×10^8 lines of flux per second.

Since the bar cuts all of the flux at each revolution and since the speed is known, we may calculate the flux or total number of magnetic lines of force necessary in the field.

Thus with N' revolutions per second, and since

$$e = \frac{\Phi \times N}{10^8}$$

where

e = voltage per conductor

and Φ = total flux-lines,

we have $$\Phi = \frac{e \times 10^8}{N'}$$

Substituting

$$\Phi = \frac{50 \times 10^8}{N'}$$

With the value of flux determined, the next step is to assume a flux density, which with steel may be from 80,000 to 100,000 lines per square inch, and determine the least diameter of a round section to carry one-half of the total flux. This determines the shaft diameter.

The diameter of the rotor proper will be somewhat larger to allow suitable space for end connections to conductors and rings, but the diameter must be such that the peripheral speed at the brushes shall not exceed 18,000 to 19,000 feet per minute.

Having found the rotor diameter, the length of rotor and stationary pole piece may be determined by finding the air gap area to carry the flux Φ. The gap density will be governed to some extent by the available space for field winding since the controlling reluctance for the magnetic circuit lies in the air gap, an average value may be said to be 60,000 lines per square inch with an air gap of say 3/8 inch.

The length of the air gap is the controlling feature of the field winding and theoretically the gap should be as small as possible. However, practical reasons such as the impossibility of absolute accuracy and balance at high speed of rotating members and the extremely heavy strains brought to bear with a small gap when the rotor is slightly eccentric, make it necessary to use a comparatively large gap.

With the air gap finally settled, the machine maybe laidout on the drawing board since the principal dimensions

have been determined with the possible exception of finding the shaft length. This may be approximated by allowing for a collector ring at each end of each conductor. From the completed sketch, the approximate lengths of the various paths of flux may be measured and the corresponding ampere turns of magnetomotive force taken from a steel magnetization curve at the assumed flux densities. The total M. M. F. may then be determined and the field coils calculated on this basis.

The output is still undetermined, but may be calculated after obtaining the diameter of the rotor, by assuming a value of ampere conductors per inch of rotor periphery which is an experience constant and has a value approximately 300 to 400. The total ampere conductors for the rotor may thus be determined, and by dividing by the number of conductors we have the current output of the machine. Knowing the terminal voltage, the output is determined and the value obtained is approximately the maximum operating value of output for this machine.

The size of parts cannot be reduced for a smaller output, since the flux and voltage would thus also be reduced and the machine would not operate properly. The only change could be a reduction in size of the conductors, but this would be undesirable from the buyer's standpoint, since at a slight additional expense above the underrated machine, he could secure the normal maximum output and with no increase in weight of steel.

The general method of generator design as indicated above gives the theoretical output neglecting all losses of friction heating, etc., but as the generator efficiency is usually quite high - about 96 percent in the larger machines - the output as calculated is approximately correct.

The calculations indicated may be simplified by the use of curves as will be shown later in the design of a 1200 volt machine operating at 750 R. P. M. The output will be approximately 5000 KW.

III GENERAL METHOD OF STEAM TURBINE DESIGN

The general method of steam turbine design furnishes a striking example in contrast with that of the acyclic generator in the method of attack, since the turbine design follows more nearly the usual methods of design in which we know the output and operating conditions and from them determine the size of parts and operating characteristics. We are, in fact, dealing with energy in the form of heat, rather than mechanical or electrical energy, and the function of the turbine will be to transform the maximum amount of this heat energy into mechanical energy of rotation. In this form it will be transmitted to the generator and there changed into electrical energy.

Before taking up the turbine design, it will be of

interest to consider the various changes or transformations of energy taking place in this turbo-generator unit with the maximum efficiency which we may expect in each case. This will provide a standard, or a theoretical maximum toward which the design will be directed and by which operating characteristics of the completed machine may be judged.

Starting with the heat energy delivered to the turbine in the form of steam at a given pressure range and superheat, we may determine the heat units entering the turbine per pound of steam supplied as,

$$H_1 = q + r + C_p\ (T_1 - T_s) \quad \text{B.t.u.}$$

at initial conditions of steam, measured above 32 degrees F. where

q = heat of the liquid (above 32° F.) at a temperature corresponding to the initial pressure p_1.

r = latent heat of evaporation.

C_p = specific heat at constant pressure of superheated steam.

T_1 = temperature of steam entering turbine.

T_s = temperature of saturated steam at same pressure p_1.

This value of initial heat can be more accurately and more easily obtained from a reliable steam table than from the equation, since the value of C_p varies slightly with temperature or degree of superheat.

This initial value of heat is not an absolute heat

measurement, since it is measured above 32° F. and not from the absolute zero of temperature. Calculations in practice, however, are usually based upon the standard steam tables, and the only quantity affected is the absolute thermal efficiency which is seldom used in practical work. Of the initial value of heat supplied in a well designed turbine station, the boiler room delivers about 80 percent of the fuel heat value to the turbines distributed among the various losses and units as follows:

High pressure steam pipes	1.6	percent
Rotation losses	.9	"
Electrical output	15.6	"
Cooling water in condensers	61.4	"
Heating feedwater	.5	"
	80.0	"

On the basis of heat supplied to the turbine, omitting piping losses, we have the following percentages:

Rotation losses	.9 percent of total.	1.2 percent of heat to turbine
Electrical output	15.6 percent of total.	19.9 percent of " " "
Cooling water	61.4 percent of total.	78.3 percent of " " "
Heating feedwater	.5 percent of total.	.6 percent of " " "
	78.4 percent of total.	100.00 percent ot turbine heat.

These values are based upon present plants using 10,000 KW units and alternators. The efficiency of an acyclic generator will be practically the same as that of an alternator, and we may thus expect these same values to represent approximately the conditions in the present design.

The apparently low efficiency of the adiabatic process is a fundamental thermodynamic condition which separates energy in the form of heat from energy in a mechanical or electrical form, thus making a distinction between "low grade" heat energy and "high grade" mechanical or electrical energy. It is a significant fact that mechanical and electrical energy are convertable practically without loss, each to the other form and both to the lower grade heat energy without any loss whatever. However, when we attempt to reverse the process, we find that the efficiency of transformation is very low, as indicated by the Carnot cycle which defines the theoretical limit of transformation in the case of a perfect gas as,

$$\eta = \frac{T_1 - T_2}{T_1}$$

T_1 = temperature of supply or heat source.

T_2 = temperature of exhaust or refrigerator.

In the case of steam turbines operating on the Rankine cycle, the form of the efficiency equation is,

$$\eta = \frac{Q_1 - Q_2}{Q_1} \text{ as a theoretical limit.}$$

or

$$\eta = K \frac{Q_1 - Q_2}{Q_1} \text{ as a practical value.}$$

where

Q_1 = initial heat per pound of steam supplied.

Q_2 = final heat per pound of steam exhausted.

K = experience constant to cover friction, churning and all rotation or radiation losses occurring in the turbine.

A practical example will probably illustrate this efficiency more clearly. Assuming operating conditions as found in modern power plants, we have.

Initial pressure	= 220 pounds per square inch absoute (or 205 pounds gauge).
Superheat	= 200° F.
Vacuum	= .75 pound per square inch (or 28.5 inches of vacuum)

From a Mollier or heat-entropy diagram, we have

Initial heat	= Q_1	= 1310 B.t.u. per pound of steam.
Final heat at same entropy and .75 pounds pressure	= Q_2	= 910 B.t.u. per pound of steam.
$Q_1 - Q_2$		400 B. t. u.

$$\eta = \frac{Q_1 - Q_2}{Q_1} = \text{maximum theoretical efficiency.}$$

$$\eta = \frac{400}{1310} = 30.5 \text{ percent}$$

The maximum value without losses is thus 30.5 percent.

Assuming a value of K = .65 to cover losses as indicated above, we have,

η' = .65 x 30.5 = 19.8 percent, which agrees very closely with the value taken from practical experience.

The remaining energy changes are those from the mechanical energy of the turbine to the electrical output of the generator, but as indicated in the previous discussion, these changes are accompanied by very slight losses as compared with those incident to the heat energy change, and will probably not exceed five or six percent.

Having thus briefly considered the various energy changes taking place in the unit under consideration, the remaining steps in the general method of design will be briefly indicated.

Having found the initial heat and entropy and having assumed the operating vacuum, the final value of the heat contents will be determined at constant entropy as indicated in the example, from the table or chart, or may be calculated. This latter process is too tedious for practical purposes and may not be as accurate as the steam tables.

The difference between the initial and final values of heat, corrected for friction and other losses, gives a value of the available energy, and from this value, we may obtain the steam consumption of the turbine:-

Available heat energy = $E_H = K\ (Q_1 - Q_2)$ B.t.u. per pound

and since 1 Horse Power hour = 2,545 B.t.u. or

1 Kilowatt hour = 3,400 B.t.u.,

we may obtain the water rate in pounds per K.W. hour (or H.P.H.) by dividing this constant by the available energy per pound of steam.

The turbine in the present case will be of the Curtis few-stage impulse type and the blade arrangement will be based upon the manufacturers' experience with the various combinations of pressure and velocity stages. After assuming the number of pressure stages and velocity stages per pressure stage the method of determining steam velocities, blade angles and nozzles may be outlined as follows.

The available heat drop is divided approximately in equal parts corresponding to the number of pressure stages so that the work per pressure stage may be practically constant. The steam velocity from the nozzles with this value of heat drop may be determined from the assumption that the change is adiabatic and that all of the heat drop appears as kinetic energy assuming initial steam velocity as zero.

$$778\ H_1^1 = 1/2 \times \text{mass} \times \text{velocity}^2$$

$$778\ H_1^1 = 1/2\ \frac{w^2}{g}$$

where H_1^1 = heat drop or available energy first stage.

w = velocity - feet per second.

g = acceleration of gravity,

= 32.2 feet per second per second.

w = 64.4 $H^1 \times 778$

$$= 223.7 \quad H_1^{1}$$

This equation gives the value of steam velocity at its exit from the nozzles if they be properly designed to avoid internal oscillations usually caused by improper nozzle length or divergence. The nozzle design is best shown by an example and will thus be omitted in this section and included in the actual design later.

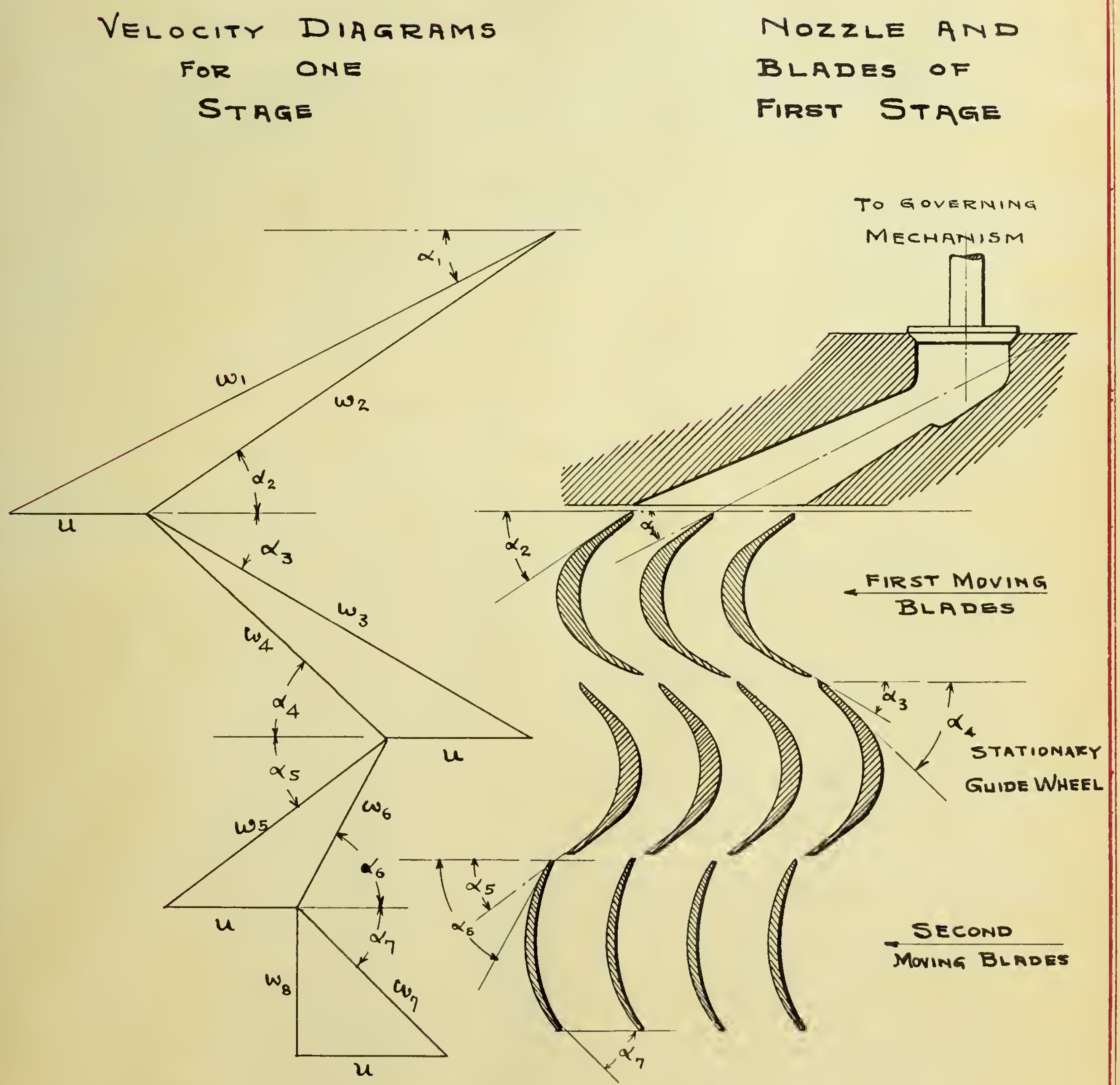
Having determined the initial velocity in the first pressure stage, a suitable blade or wheel speed is chosen on the basis of an assumed number of velocity stages for this pressure stage. The peripheral velocity will also be limited by mechanical considerations, and several assumptions may be necessary before a satisfactory arrangement is obtained. The number of velocity stages in the example used previously, (Page) might be found as an example of this method.

$$.65 \ (H_1 - H_2) = \text{available heat drop} = 260 \text{ B.t.u. per lb.}$$

Assuming five pressure stages, the heat drop per stage will be 52 B.t.u.lb. The initial velocity will then be

$$w_1 = 223.7 \quad 52$$

= 1610 feet per second, or say 1500 feet per second is the speed component parallel to the direction of the rotating buckets. With the velocity stages or wheels for each pressure stage, and assuming a wheel speed of say 350 feet per second, the blade angles may best be found graphically as indicated in Figure 5, page 19. Neglecting the effect of friction in the blades, an approximate determination



VELOCITY DIAGRAMS FOR ONE STAGE
NOZZLE AND BLADES OF FIRST STAGE
TO GOVERNING MECHANISM
FIRST MOVING BLADES
STATIONARY GUIDE WHEEL
SECOND MOVING BLADES
α_1
α_2
α_3
α_4
α_5
α_6
α_7
w_1
w_2
w_3
w_4
w_5
w_6
w_7
w_8
u

FIG. 5

of blade angles may be made by laying the velocities out to scale and measuring the angles (See Figure 5). Thus ω_1 is the abolute initial velocity at the angle α_1 and when the blade velocity u is vectorially subtracted, the angle ω_2 is obtained as the relative velocity of the jet with respect to the blades and α_2 is thus the blade inlet angle. The steam is then reversed in the first moving blades to the direction ω_3 and leaves the blades at the absolute velocity ω_4. The angle α_4 is then the inlet angle for the first set of stationary blades and ω_5 is the absolute steam velocity entering the second velocity stage. The other angles follow in the order indicated in the sketch, reducing the axial component of the steam velocity to as low a value as possible and leaving the final velocity ω_8 from the last set of moving blades, practically parallel to the turbine axis.

The steam then passes to the second stage through another set of nozzles and acquires kinetic energy, as before, by the heat drop accompanying a drop of pressure in the nozzles. Another series of velocity stages similar to those in the first stage but of larger size is provided to extract the kinetic energy of the steam and the process continued until the last stage has been reached at condenser pressure.

During this adiabatic process, the quality of the steam necessarily decreases and will probably reach saturation values although the tendency of all rotational and leakage losses is to increase the quality.

The final quality allowing no losses in the turbine example used, would be 82 percent, while the 35 percent fric-

tion losses increase the quality to 96 percent and the entrophy from 1.66 to 1.92 showing the increase of unavailable energy due to the rotational losses.

The design of turbine casing, wheels, shaft, diaphrams and other parts will be indicated in the calculations and under the heading of "Mechanical Problems".

A photo-reduction of Marks and Davis - Total Heat - Entropy chart is appended to aid the reader in following the example given, as well as in the design calculations following. (See Figure 6).

IV. ELECTRICAL CALCULATIONS AND DESIGN

The assumed conditions of operation for the acyclic generator must conform, especially in the question of speed, with the suitable conditions of turbine operation in order that the unit as a whole may operate most efficiently. The speed of a 5,000 K.W. turbine of the Curtis type should be approximately 750 R. P. M. and the generator will, therefore, be designed on this basis.

With a definite speed determined, it is only necessary to know the total voltage and the voltage per bar in order to proceed with the calculations. This unit is to operate at 750 R. P. M., and to develop a total E. M. F. of 1200 volts. It is expected that the output will be approx-

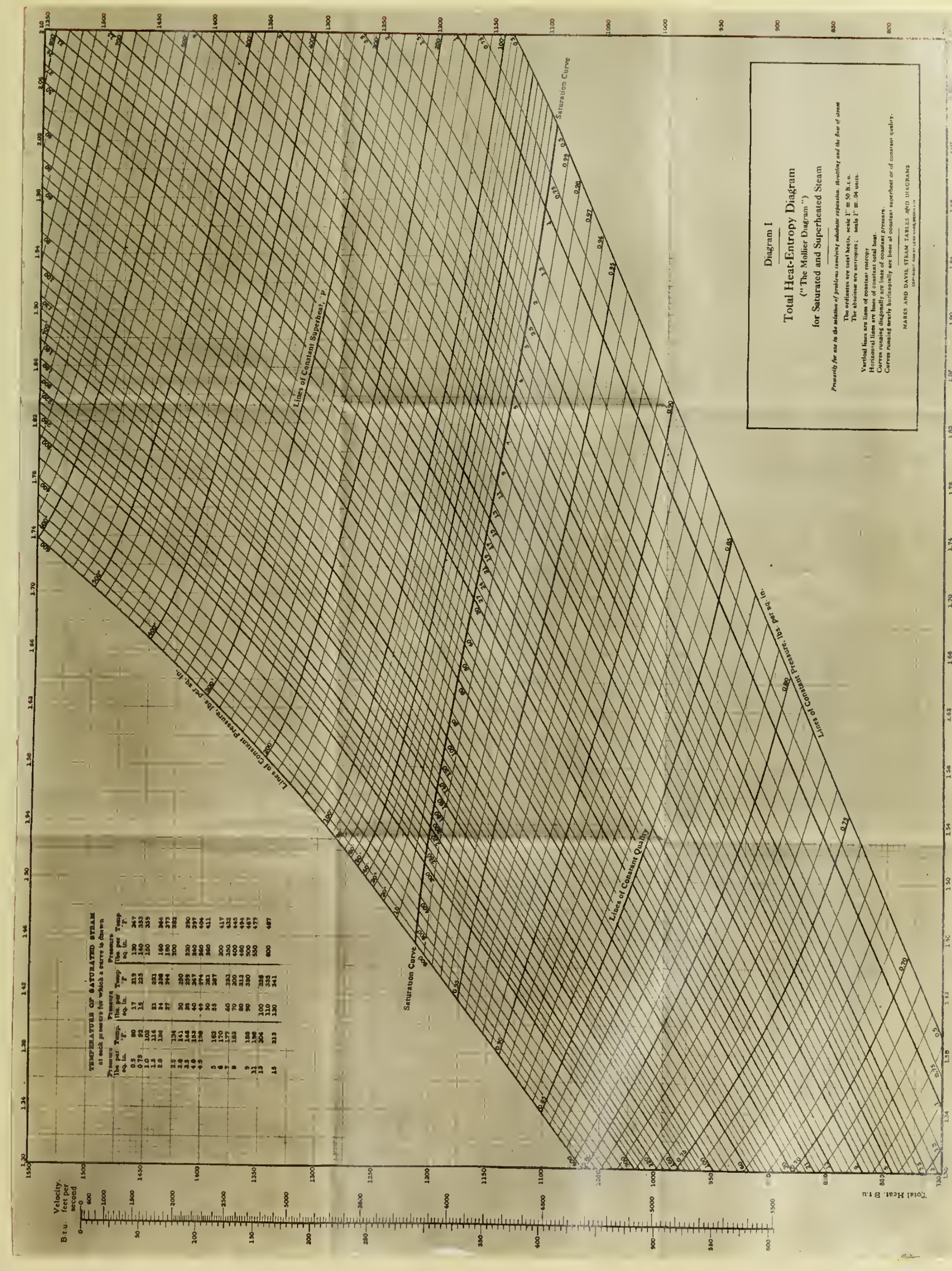


Diagram I
Total Heat-Entropy Diagram
("The Mollier Diagram")
for Saturated and Superheated Steam
Lines of Constant Superheat, F°
Lines of Constant Pressure, lbs per sq. in.
Lines of Constant Quality
Saturation Curve
TEMPERATURES OF SATURATED STEAM
Total Heat, B.t.u.
Velocity, feet per second
B.t.u.

FIG. 6

imately 5,000 K. W., but this value will be determined later, as indicated in the general method.

It will be convenient in the calculations to use the number of generated volts per conductor as a basis from which other quantities may be determined. Furthermore, it will be found more satisfactory to calculate various values of constants leading to the value of output, for different assumptions of volts per conductor and to plot them in curve form, than to attempt all determinations by calculation. The curves will also indicate more clearly the effect of altering various constants and will indicate the practical limitations imposed by size, weight centrifugal force, peripheral speed and so forth.

Calculations.

The values of volts generated per armature bar are arranged on the first data sheet (Page 32) from a value of 30 to one of 90 volts per conductor.

Let e = voltage per conductor

and E = total voltage.

N = number of conductors in series (which includes all conductors in this machine).

$$E = ne \tag{2}$$

$$n = \frac{E}{e}$$ plotted in the second column.

Since

$$e = \frac{n \Phi}{60 \times 10^8} \tag{3}$$ where n = revolutions per minute

Φ = total flux as before.

we have flux

$$\Phi = \frac{e \times 60 \times 10^8}{n} \tag{4}$$

taking the first value of 30 volts per conductor

$$\Phi = \frac{30 \times 60 \times 10^8}{750} = 240,000,000 \text{ or } 240 \text{ megalines.}$$

Having determined the flux, the area of the smallest section will be that sufficient to carry one-half of this flux at a density of say 80,000 lines per square inch. This assumption is based upon using the type of generator shown on page , Figure .

The least shaft diameter which carries the flux may be determined and the value of peripheral speed obtained. Thus in the first case it is found that with 240 megalines total flux, 120 will be carried at each end of the shaft and at 80,000 flux density this gives 1500 square inches area and 43.7" in diameter. The peripheral speed will be 8,610 feet per minute with 750 R. P. M. This is well within a limiting value of 19,000 feet per minute.

Assuming an air gap density of 60,000, the necessary area to carry the total flux is 4,000 square inches while at 80,000 it is 3,000 square inches.

The conductors will be set in holes drilled the length of the solid rotating pole piece near the periphery and the necessary added diameter to allow for these conductors and also their collector ring connections, will be about four to seven inches depending upon the current value in the conductors. Having found the rotor diameter, its length will be determined by the air gap area as found above. The periphery of the rotor in this case is 50 x 3.1416 = 157 inches, or taking a mean circumference in the 3/8 inch airgap, it will

be approximately 158 inches. The air gap area at a density of 80,000 lines per square inch was found to be 3,000 square inches. The rotor length will thus be 3,000 divided by 158 or 19 inches. In the same manner, the lower density of 60,000 lines gives the rotor length as 25.5 inches and illustrates the effect of varying magnetic density upon the size and weight of steel employed.

The air gap is assumed as 3/8 inch which may seem large at first thought, although when the effect, with a small gap, of a slight eccentricity in the bearings is considered, the reason is more apparent. With a high air gap density and small gap a slight variation of the rotor position would cause an excessive variation of flux density resulting in hysteresis losses and unequal voltage generation and at the same time, causing a disastrous side pull on the bearings. With a larger gap the possible percentage of variation in the gap length is reduced with corresponding reduction of the effects noted.

At 400 ampere conductors per inch of periphery, a 50 inch diameter would allow a total of 62,840 ampere-conductors. With 40 conductors on the rotor, this gives a current value of $\frac{62,840}{40} = 1,570$ amperes (5)

The output may then be directly determined as

$$K.W. = \frac{1,570 \times 1200}{1,000} = 1,882 \text{ Kilowatts} \qquad (6)$$

The bar diameter is determined by the value of current density per square inch. Current practice for well

ventilated rotating members is 2000 amperes per square inch, but due to the enclosed nature of this generator, a value of 1500 amperes per square inch will be assumed as a basis of calculation. This will enable the machine to carry more than the rated current without excessive heating, which is often an advantage in this type of unit as the steam turbine is capable of carrying heavy overloads and thus the generator should be designed accordingly.

As a check on the mechanical construction, the clearance between adjacent bars is calculated to make certain that the steel area is sufficient both to carry the flux and to withstand centrifugal forces. In the first calculation at 30 volts per conductor, the bar diameter is 1.15 inches and the clearance between adjacent bars is 2.6 inches. The centrifugal stress neglecting the strengthening effect of the metal outside of the conductor may be found as follows:-

Taking one inch as a unit length of rotor for the stress calculation, we find from a sketch of the bar and rotor spacing (Figure 7) that the space between the bars is

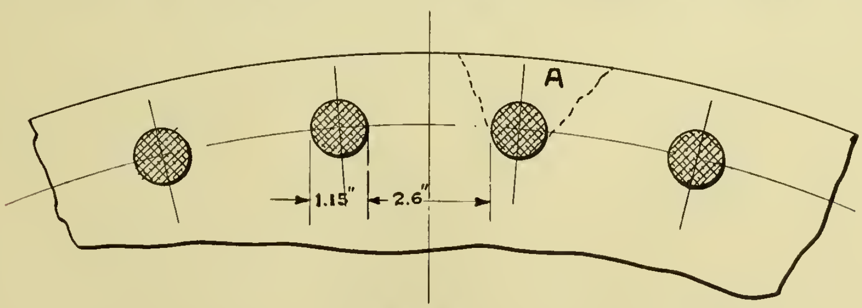


Figure 7.

too large to be the controlling factor of centrifugal force. The fracture due to centrifugal force of the bars would probably follow a line "A" and should therefore be calculated

on the basis of shearing stress.

Assuming the length of the fracture "A" as a minimum of 1.3", there is a total shear-resisting area of 2.3 square inches per inch length of rotor per conductor.

Centrifugal force

$$F_c = M \frac{v^2}{r} \quad \text{where } M = \text{mass}$$

v = speed of mass in feet per second

r = radius to mass center

$$F_c = \frac{w}{g} \quad \frac{v^2}{r} \qquad (7)$$

w = weight of conductor pounds per inch length

g = acceleration of gravity

Substituting w = 3.08 lb. v = 156 ft. per second,

$$F_c = \frac{3.08}{32.2} \times \frac{24,400}{23.8} = 99 \text{ pounds.}$$

The centrifugal force due to the conductor is thus about 100 pounds per inch of length. To this is added the centrifugal force of the wedge-shaped block of steel. In this case the weight w is approximately that of 1-1/2 cubic inches of steel or about .5 lb.

$$F_3 = \frac{.5}{3.08} = 99 = 16 \text{ pounds.}$$

The total centrifugal force is then approximately 115 pounds per inch and the shearing stress is practically negligible, being $\frac{115}{2.3}$ = 50 pounds per square inch.

The conductor might be placed nearer the periphery but this would weaken the collector rings near their point

of wear and would thus be inadvisable. The question of armature reaction also enters, and is considered later.

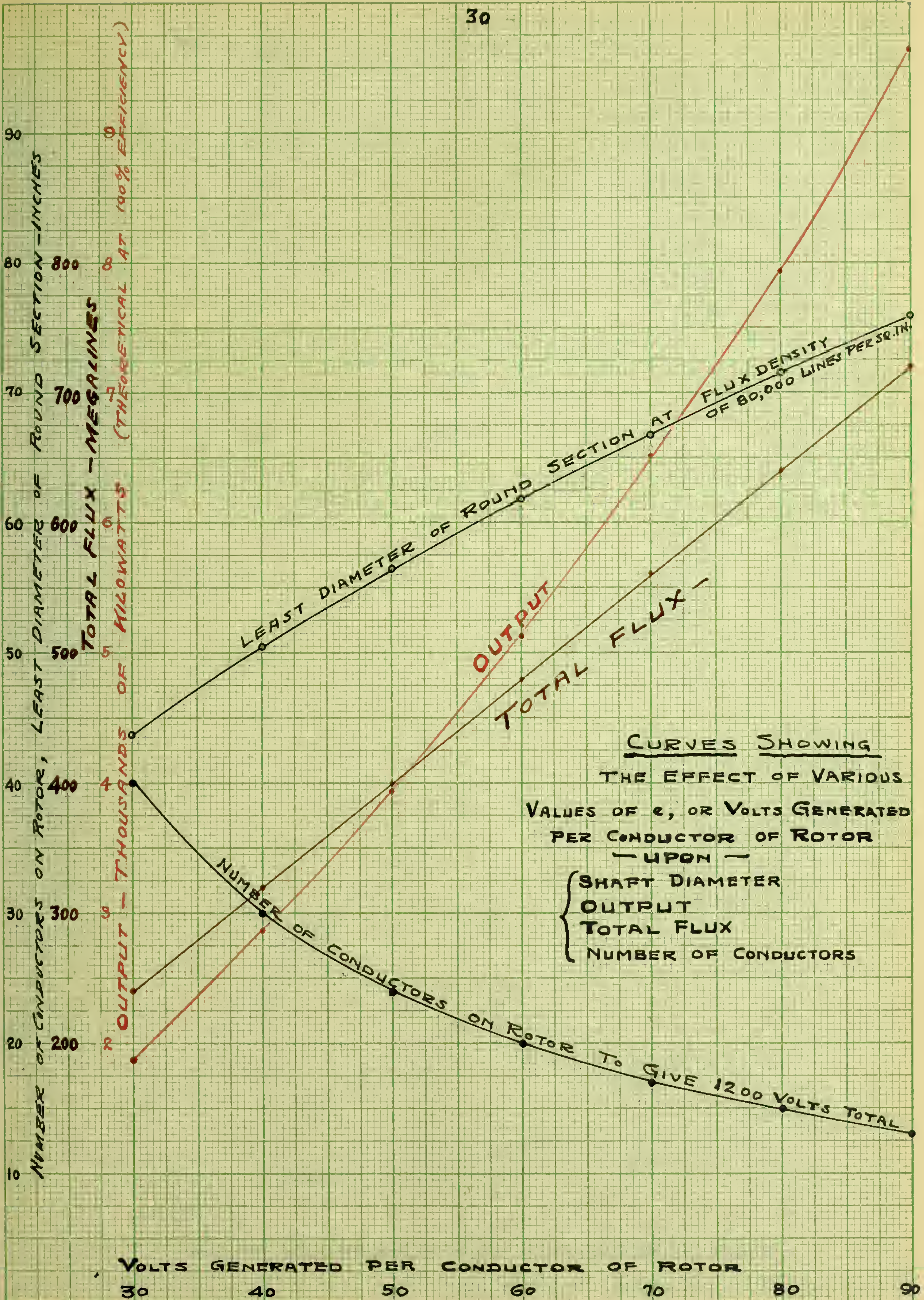
From the curves it is apparent that to obtain 5,000 K. W. with 750 R. P. M. and 1200 volts pressure that it will be necessary to use approximately 59 volts per conductor which is rather a high value and indicates that either the speed assumed was too low or else the voltage was too high.

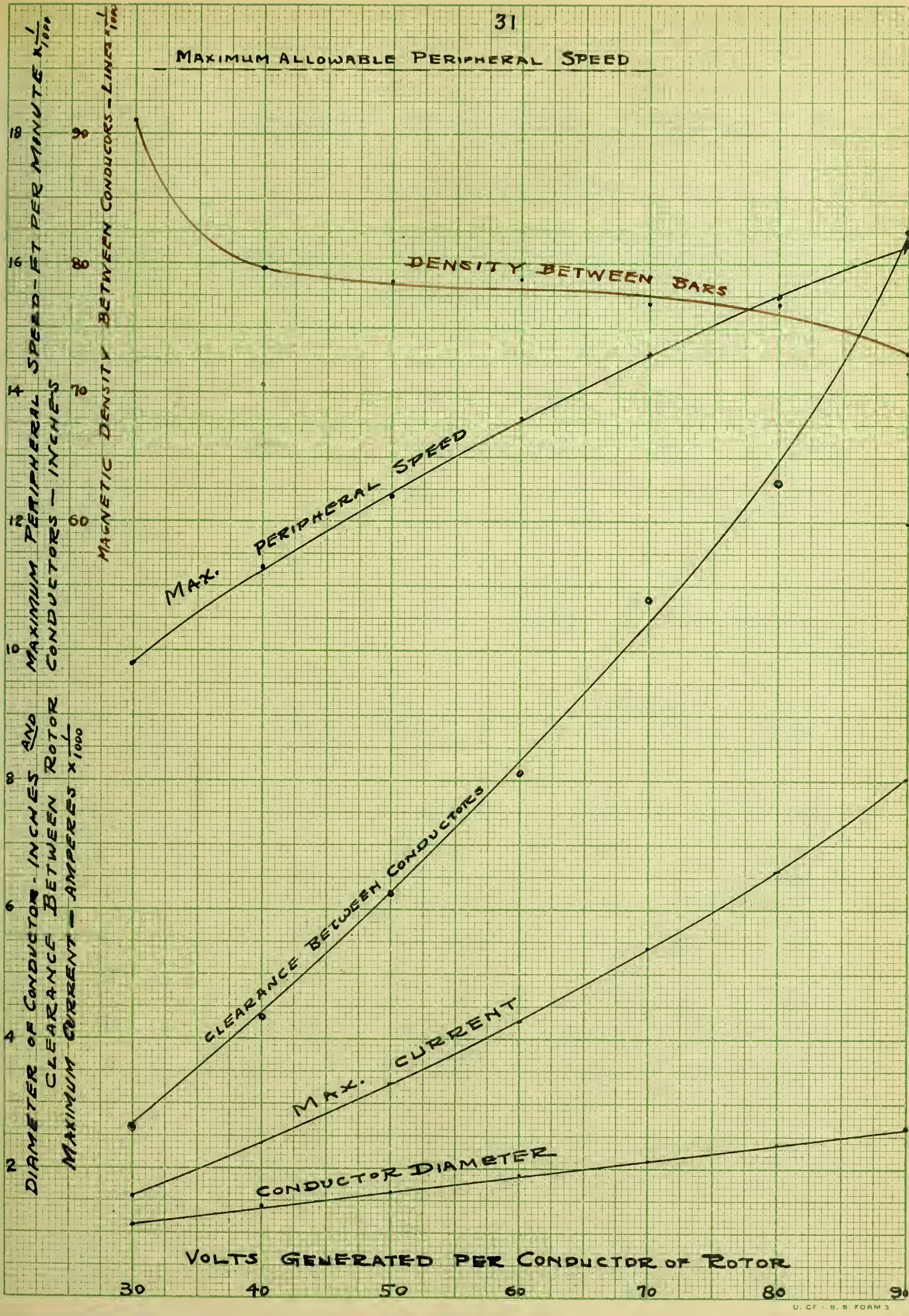
For the sake of having an even number of conductors on the rotor, the field coils will be calculated on the basis of 60.0 volts per conductor with 20 conductors in series, giving approximately 1200 volts at no load.

Since all fields are practically constant, it will be unnecessary to use laminated construction and the only condition thus imposed upon the field and rotor parts is that they be homogeneous, or, that if they are made of cast steel, the castings must be free from blow holes or other imperfections.

Before determining the field excitation it will be necessary to study the effect of the secondary or rotor magnetization upon that of the primary or field. There are two sources of reaction in the rotor, namely, the reaction of the conductors and the magnetic effects of the collector rings which obviously carry pulsating currents and thus set up pulsating magnetomotive forces.

Of these reactions the first is the most important, although the second requires careful consideration in balancing or neutralizing the ring reactions. The conductor, or armature, reaction cannot be balanced since it is uni-direc-


Curves Showing
The Effect of Various
Values of e, or Volts Generated
Per Conductor of Rotor
— Upon —
Shaft Diameter
Output
Total Flux
Number of Conductors
Least Diameter of Round Section at Flux Density of 80,000 Lines per sq. in.
Output
Total Flux -
Number of Conductors on Rotor to Give 1200 Volts Total
Number of Conductors on Rotor, Least Diameter of Round Section - Inches
Total Flux - Megalines
Output - Thousands of Kilowatts (Theoretical at 100% Efficiency)
Volts Generated per Conductor of Rotor
30
40
50
60
70
80
90
10
20
30
40
50
60
70
80
90
200
300
400
500
600
700
800
2
3
4
5
6
7
8
9



MAXIMUM ALLOWABLE PERIPHERAL SPEED
DIAMETER OF CONDUCTOR - INCHES AND MAXIMUM PERIPHERAL SPEED - FT PER MINUTE x 1/1000
CLEARANCE BETWEEN ROTOR CONDUCTORS - INCHES
MAXIMUM CURRENT - AMPERES x 1/1000
MAGNETIC DENSITY BETWEEN CONDUCORS - LINES x 1/1000
DENSITY BETWEEN BARS
MAX. PERIPHERAL SPEED
CLEARANCE BETWEEN CONDUCTORS
MAX. CURRENT
CONDUCTOR DIAMETER
VOLTS GENERATED PER CONDUCTOR OF ROTOR
2
4
6
8
10
12
14
16
18
60
70
80
90
30
40
50
60
70
80
90

VARIATIONS DUE TO CHANGE OF CONSTANT

VOLTS PER CONDUCTOR

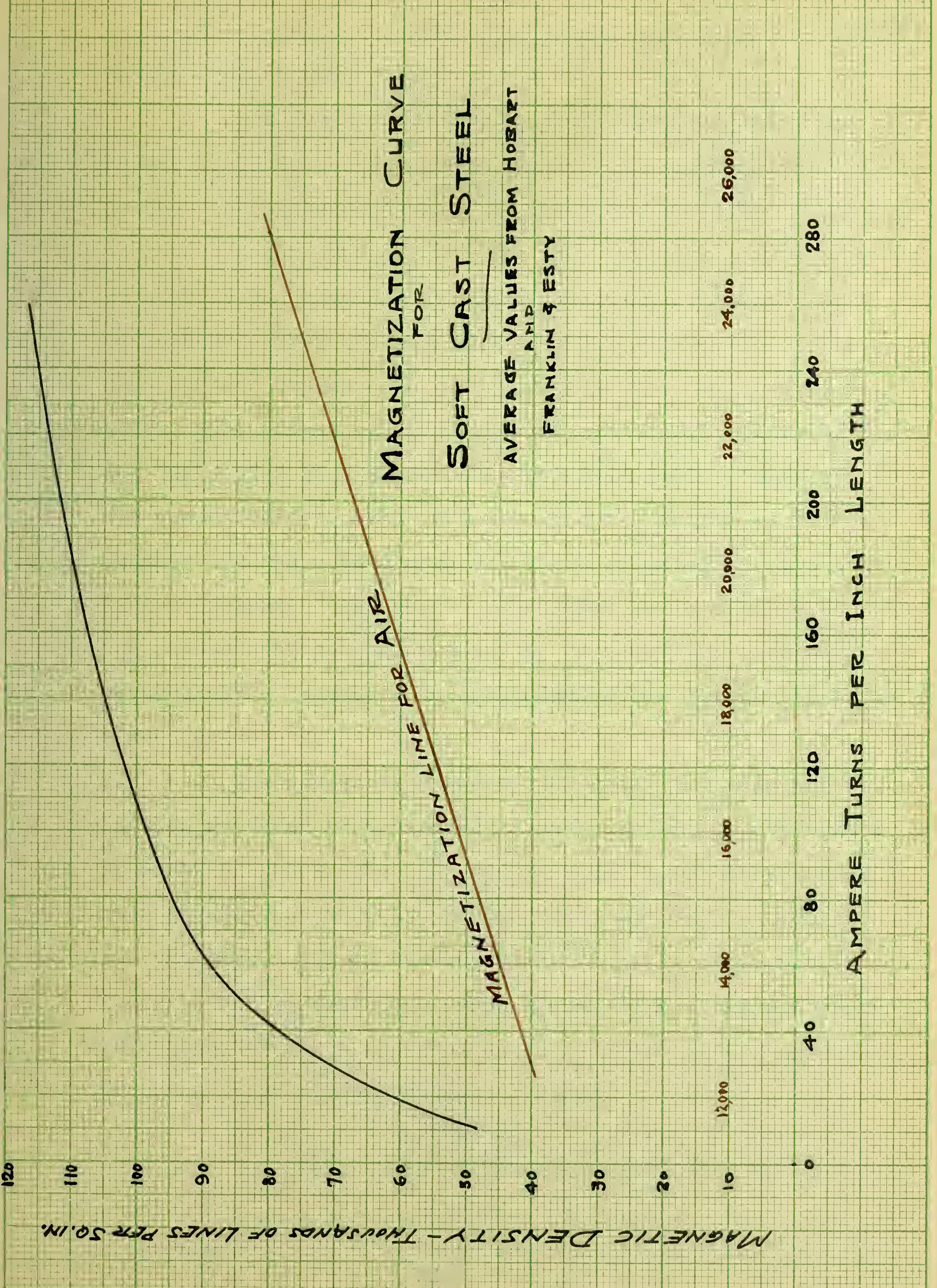
	Volts Cond.	Number Conductors	Total Flux Φ Megalines	Least Area for $\frac{\Phi}{2}$ at 80.000	Least Diameter Round Section	Air Gap Area at 60,000
1	30	40	240	1500	43.7	4000
2	40	30	320	2000	50.5	5340
3	50	24	400	2500	56.5	6670
4	60	20	480	3000	61.8	8000
5	70	17	560	3500	66.8	9340
6	80	15	640	4000	71.4	10670
7	90	13	720	4500	75.8	12000

	Approx. Rotor Diameter Inches.	Length Rotor Inches	Amp. Cond. at 400 per inch.	Output amp.	Output K.W.	I Max.
1	50	25.5	62,840	1,570	1,582	1,570
2	57	31.8	71,800	2,395	2,870	2,395
3	63	33.7	79,100	3,300	3,960	3,300
4	69	37.5	86,600	4,330	5,200	4,260
5	74	40.1	93,000	5,460	6,560	5,400
6	79	43.0	99,100	6,600	7,930	6,600
7	83	46.0	104,200	8,020	9,640	8,020

VARIATIONS DUE TO CHANGE OF CONSTANT

VOLTS PER CONDUCTOR

Volts / Cond.	Bar Area at 1500 am./sq.in.	Bar Diam. In.	Circum. Bar Circle $d = D_R -$	Clearance Between Bars	Mag. Den. Between Bars	Max. Per. Speed Ft. Per Min.
30	1,05	1.15	150	2.6	91,000	9,800
40	1.60	1.43	173	4.34	79,600	11,200
50	2.20	1.67	190	6.25	78,600	12,400
60	2.84	1.9	208	8.1	78,800	13,600
70	3.60	2.14	219	10.8	76,900	14,600
80	4.40	2.36	238	12.6	77,000	15,500
90	5.35	2.61	249	16.5	73,[illegible]00	16,300



U. OF I. S. S. FORM 3

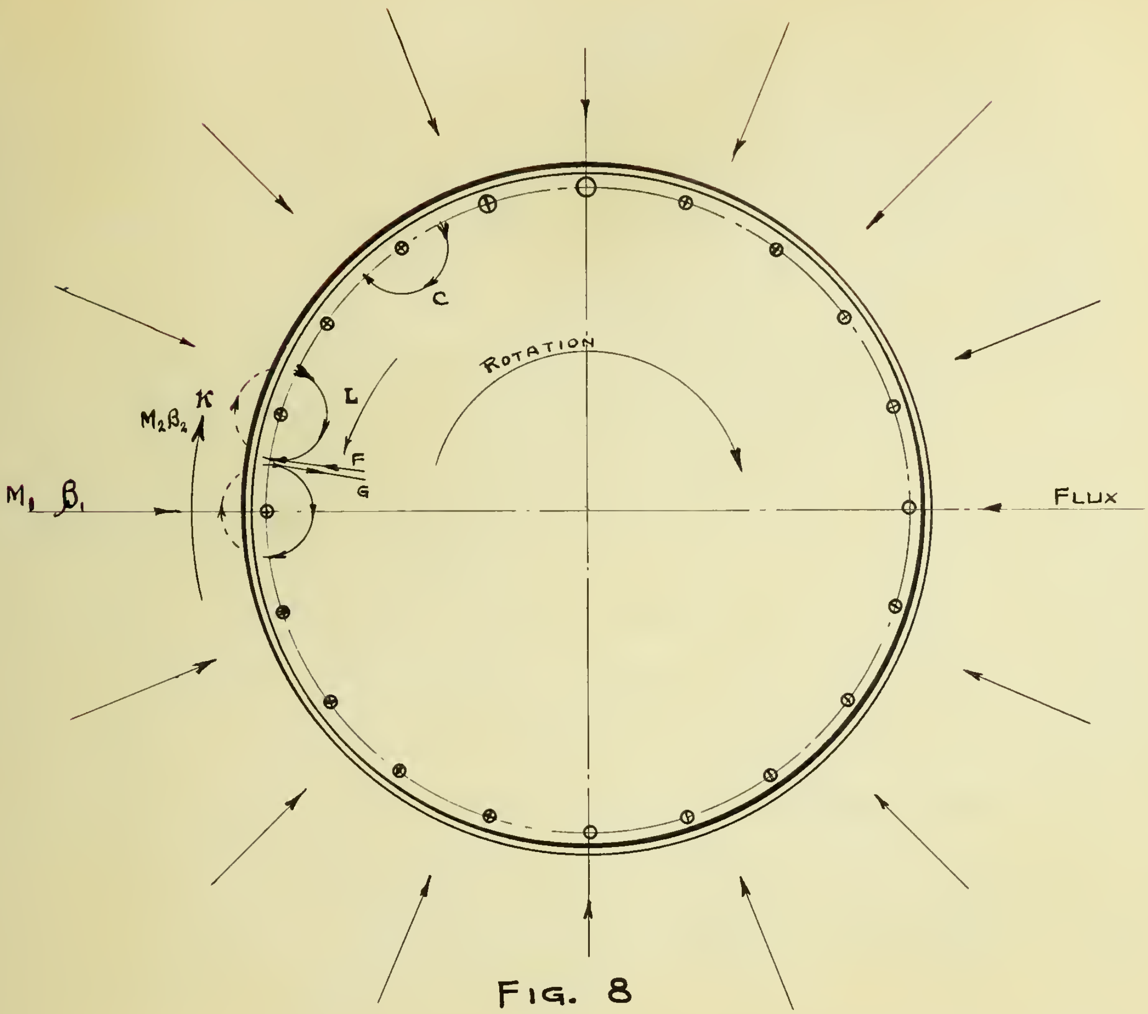
ectional and must, therefore, have an effect upon the magneto-motive force of the field.

Referring to Figure 8, it is apparent that the conductor reactions are cross-magnetizing and thus tend to distort the paths of flux which are normally radial. Assuming the stationary field pole as North and the rotation as clockwise, the armature conductors will carry current away from the observer as indicated by the arrows. The magneto-motive force around each conductor will then be clockwise as at c. The effect of this magnetic action will be to oppose the field flux at F and to strengthen it at G. However, the magnetomotive forces F and G tend to neutralize one another and the effect upon the field or primary reluctance is very slight. The reaction may thus be resolved into two magneto-motive forces K and L, of which the latter may be neglected, leaving for consideration the element of cross magnetization, K.

The effect of armature reaction, or cross magnetization may best be indicated by a vector diagram, Figure 9, in which M_1 and B_1 are the primary magnetomotive force and flux density respectively in the field pole and M_2 and B_2 are the magnetomotive force and flux density produced by the armature conductors. Then M_R and B_R are the resultant magnetomotive force and flux density providing the reluctance is constant in all directions. Then

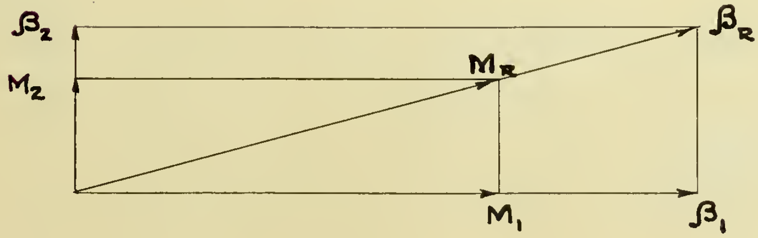
$$B_R = \sqrt{B_1^2 + B_2^2} \text{ with constant reluctance.} \quad (8)$$

In the present design, one of the objects of choos-



ROTATION
C
L
K
$M_2 \beta_2$
F
G
M_1 β_1
FLUX

FIG. 8



β_2
M_2
M_R
β_R
M_1
β_1

FIG. 9

ing a large air gap was to reduce the armature reactions, since with a large gap, the reluctance in the path of the secondary magnetomotive force is very great, being two lengths of air gap for the flux surrounding any given conductor and entering the stationary pole. The secondary flux is then so small that its effect will be inappreciable and may thus be neglected. The relation indicated in equation 8 above, due to varying reluctance will not be exact, but the component of primary magnetization will be slightly decreased and to restore it to its original value it will only be necessary to add as many ampere turns as are needed to balance the increase of density and reluctance in that part of the circuit affected by the armature magnetomotive force M_2. It is evident that this increase will be very slight and will be counter balanced by the compounding effect which may be obtained through proper arrangement of the brushes as indicated later.

In arranging the rotor windings and collector ring connections, it will be necessary to consider the following conditions:-

1. Neutralization of ring reactions.
2. Minimum pressure between rings.
3. Magnetizing effect of end connections to brushes.
4. Grouping of brushes for accessibility.

Of these operating requirements, the first is most important since with the large currents in the rings, the

effect of unbalancing would be destructive to the primary magnetization and would cause endless trouble from heating and eddy current losses as well as bad voltage regulation. The condition to be met is that Σ magnetomotive forces of the rings = 0.

The method of obtaining a zero resultant of the ring magnetomotive forces may be readily developed when we consider that each ring produces a pulsating magnetomotive force as the conductor and ring revolve. Thus in Figure 10, a balance is obtained between two rings by making bar connections on opposite sides if the brushes are placed in the same relative position. The magnetomotive force in each position is zero, since at A the current passes directly to the brush and at B the current divides equally, making a half turn in each direction. The conditions will be exactly opposite as the rings revolve and thus a balance is maintained in all positions.

The same may be said of the system shown in Figure 11, which follows directly from Figure 10. Now by rotating

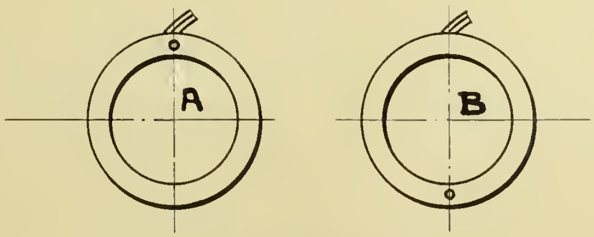


Figure 10.

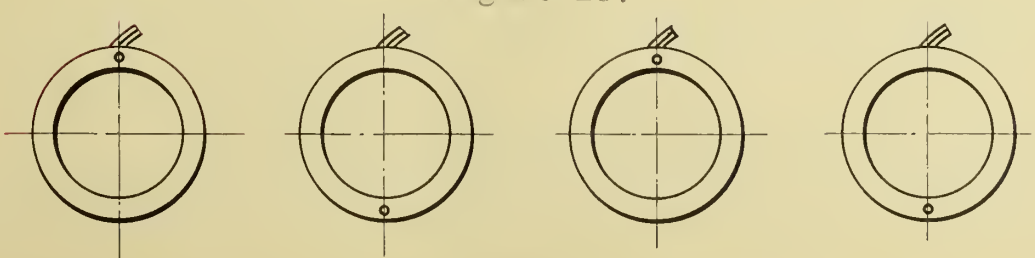


Figure 11.

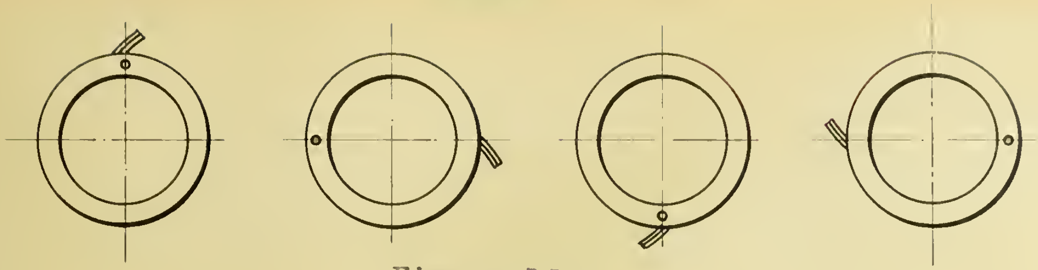


Figure 12.

the alternate brushes and conductors one-quarter turn to the right and left respectively, the condition of balance is still not changed since the time relations of the magnetomotive force pulsations have not been altered. Figure 12 indicates this condition which enables us to obtain a convenient distribution of the brushes into four groups, and still maintain balanced conditions.

By building up the end connections of balanced groups as in Figure 12, the effect of ring reactions may be entirely neutralized. A careful inspection will show that if the end connections proceed counter-clockwise, the brush connections must follow in the opposite, or clockwise direction, so that if the twenty conductors are connected to the collector rings with joints following a left hand helix, the brushes, starting in the same position will be placed at quarters and following a right hand helix. In this manner the first and fourth conditions are satisfied.

On account of equal resistance and to facilitate parallel operating, all conductors will be the same length and the end connections at one end will be the reverse of those at the other.

The spiral or helical direction of ring connections also satisfies the second requirement of minimum pressure

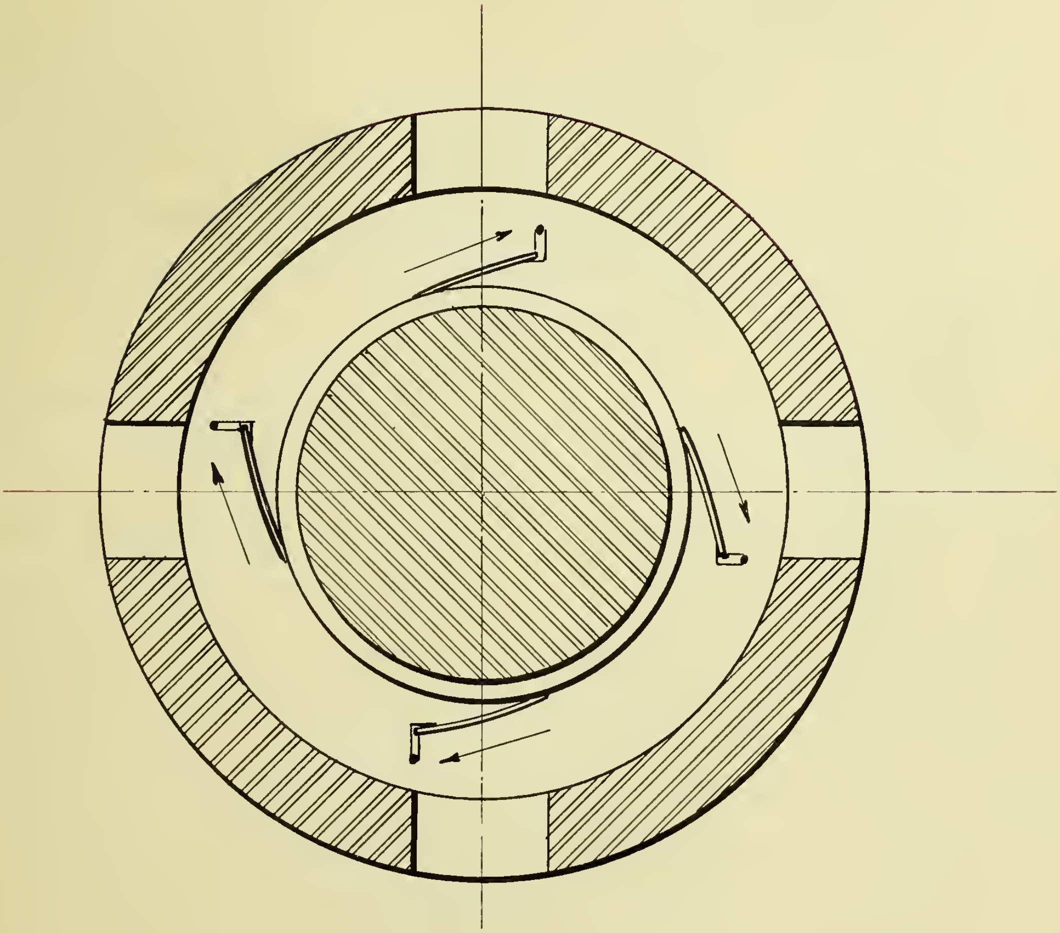


FIG. 13

between rings and it remains only to indicate the method of obtaining compounding by the use of brush connections.

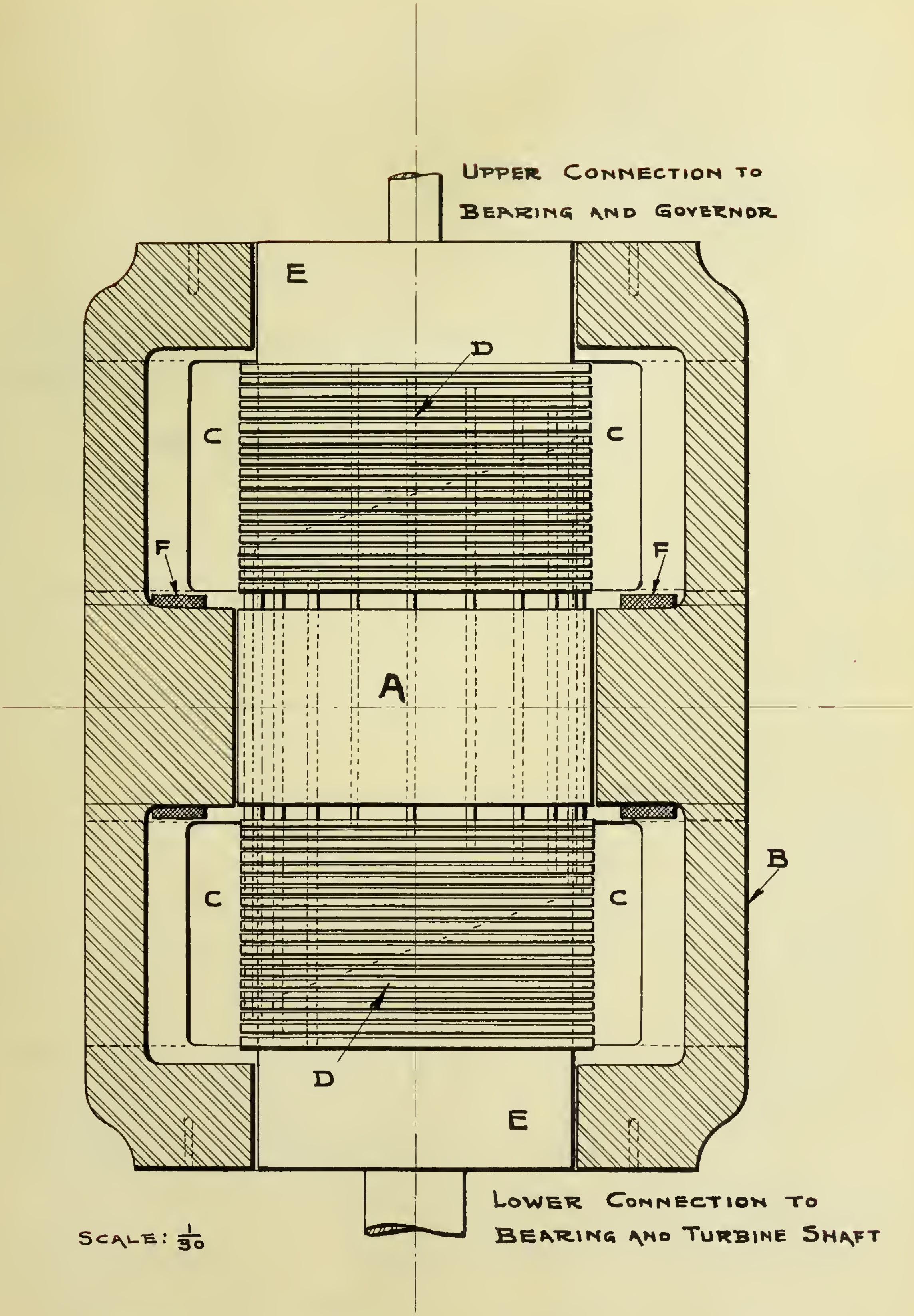
Referring to Figure 13, which shows an end view of the brush sets, it is evident that in passing from the collector ring to the cable, the current traverses the brush in a direction parallel to the field coil winding and there is thus a magnetomotive force produced which may be arranged either to aid or oppose the field coil. In thie machine the brush connections will be arranged to increase the field with increasing load and thus balance the effects of saturation and armature reaction which tend to decrease the flux and impair good regulation.

Figure 14 is an approximate drawing to a scale of 1/30, indicating the final arrangement of the acyclic generator. The mean lengths of flux paths may now be obtained and the proper magnetomotive force provided. The drawing is also arranged to indicate the vertical position of the shaft, since in this mechanical arrangement the question of bearings is greatly simplified and the entire weight of acyclic rotor and turbine wheels is carried is carried on a suitable high pressure oil step bearing located below the turbine condenser base.

Starting with the rotor diameter of say 70 inches and effective length of 37.5 inches, the remainder of the dimensions may be approximated as follows:-

Allowing 18 inches outside of rotor (A) for locating brushes, the inside diameter of the shell (B) is 106

[illegible]



Section of Acyclic Generator

Fig. 14.

inches. One-fourth of this periphery will be open for brush installation, adjustment or ventilation, making four openings, (C), each about twenty inches wide.

With a magnetic density of 90,000 lines per square inch, the area for one-half of the flux is 2,667 square inches. By computing the dimensions of a ring of 106 inches inside diameter and of area equal to 4/3 of 2,667 square inches, the metal thickness is approximately 9.5 inches

In approximating the length dimensions, 2-1/2 inches is allowed for each collector ring, making 45 inches at each end of the rotor for rings. (D) Beyond the rings at each end is an air gap through which the flux enters the shaft. (E) This gap carries one-half of the total flux and thus at a density of 60,000 lines per square inch would require a total area of 4000 square inches. The length of the gap with a diameter of 62 inches would then be 20.5 inches.

These end air gaps being located nearer the bearings will require less clearance than the main air gap and will thus be calculated on the basis of 1/4 inch, making a total of 5/8 inch air gap in the flux path. It will thus be apparent that the major part of the field magnetomotive force will be required for the air gaps.

With one inch gap at 60,000 lines per square inch the magnetomotive force necessary is 18,800 ampere turns, or with 5/8 inch gap, the field coil at each end will furnish at least 11,750 ampere turns. The field calculation is made

as follows:

Part	Density Lines per Sq. in.	A. T. per in.	Length In.	Total Ampere Turns
Rotor	80,000	42	100	4,200
Air Gaps	60,000	18,800	5/8	11,750
Pole Pieces	60,000	19	50	950
Shell	90,000	62	40	2,480
				19,380
		Leakage = 10%		1,940
	Ampere turns each end			21,320

Assuming an exciting current of 100 amperes the number of turns will be 214, and the wire size at a current density of 1500 amperes per square inch will be 1/15 or .0666 square inches.

The field coils (F) will thus have 214 turns of Number 1 B and S wire, which with a diameter of approximately .3inch including insulation will give a total coil sectional area of 19.3 square inches. The field coil could then be formed to a section 2 inches wide and 10-1/2 inches deep.

The total length of No. 1 B. and S. wire for both coils would be approximately 10,480 feet with a total resistance of 1.31 ohm. The exciting current of 100 amperes would thus be supplied at 131 volts, or from a 150 volt generator of 15 K.W. rating. The power consumed in excitation is thus .3 percent of the total rating, which compares

favorably with that of other turbine driven generators.

The efficiency of the generator may best be approximated by calculating or assuring the various values of losses encountered. Principal among these is the loss at the brushes in friction. Using steel banded rings with brushes made up of nickel steel and phosphor bronze, interleaved, the losses would be somewhat reduced from those obtained with copper brushes. Ring wear would be still further reduced by using a lubricating pilot brush of graphite on each ring which would prevent cutting at high current density. From machines previously constructed, it is found that the friction loss from each brush at 14,000 feet per minute of ring speed was approximately .5 K. W. The voltage drop at each contact at this speed and 4000 amperes would be approximately 1.6 to 1.8 volts. Assuming the lower value, on account of large ring space allowed, a voltage drop of about 64 volts is encountered.

The ring losses at full load may then be approximated as follows:-

Friction = 40 contacts at .5 K. W. each = 20 K. W.

Voltage drop - 64 volts and 4300 amperes = 275 K. W.

Resistance of conductors and cables,

300 feet of 3,620,000 cir. mils section.

Resistance = .0009 ohm and I^2R loss at 4300 amperes =

17 K. W.

Total excluding winage and hysteresis 312 K. W.

The full load efficiency on the basis of

$$\text{Efficiency} = \frac{\text{Output}}{\text{Output + Losses}}$$ will be

$$\text{Efficiency} = \frac{5000}{5312} = 94.1 \text{ percent.}$$

An approximate efficiency curve is indicated on page 47 .

The voltage regulation, without considering the compounding effect of end connections, will depend upon the contact and resistance losses:-

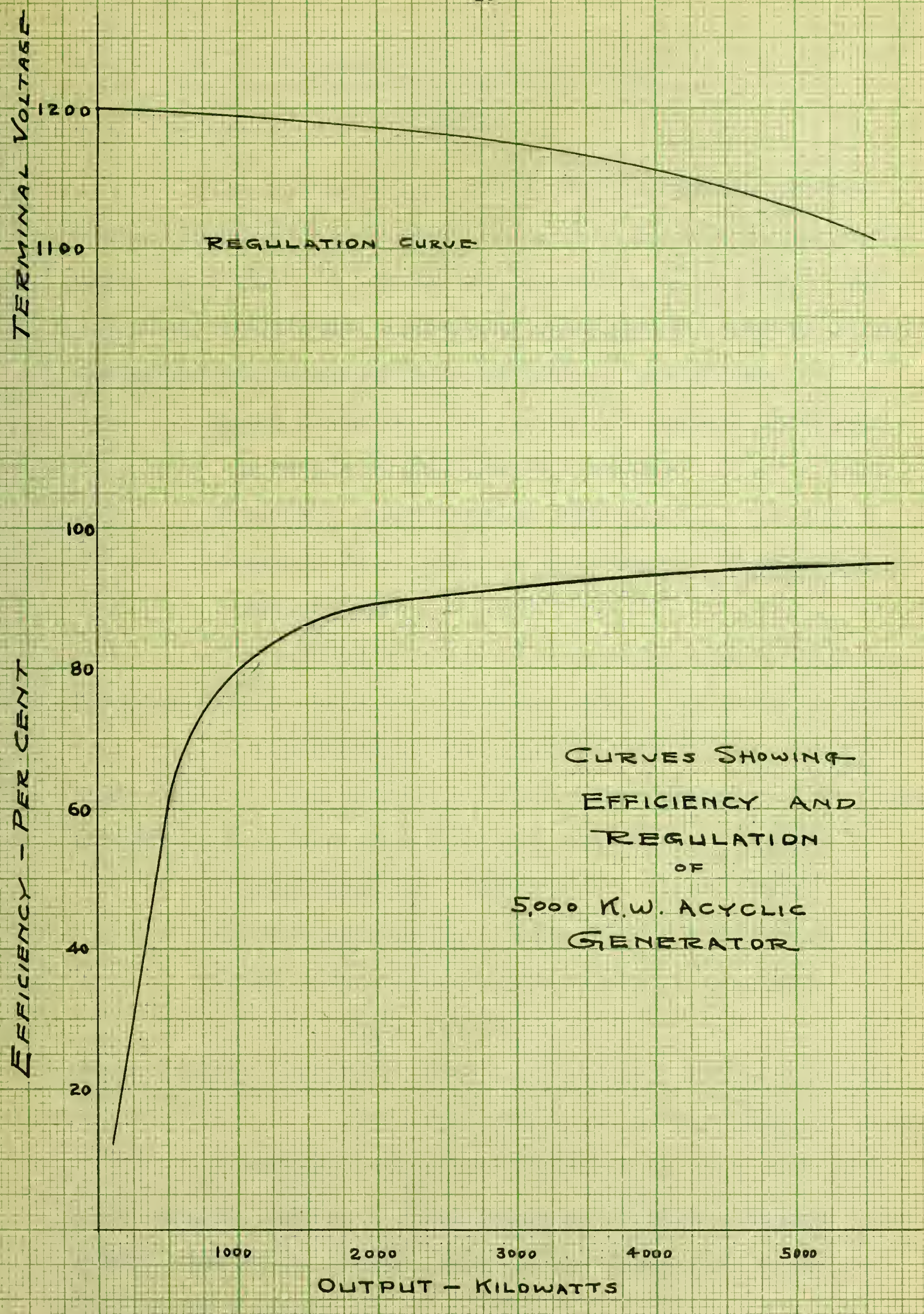
Contact loss		64 volts
Resistance drop	.0009 ohm and 4300 amperes	4 volts
		68

Assuming a total drop of 70 volts,

$$\text{Regulation} = \frac{70}{1200} = 5.8 \text{ percent - say 6 percent.}$$

An approximate regulation curve is shown on page 47.

In conclusion, it might be said that since the frictional losses remain practically constant at high speeds and since the resistance loss is a small percentage of the total, it would be fair to assume that the efficiency would increase with over-load before the increasing losses would reduce it, so that a maximum efficiency of 96 percent would probably be realized.


Terminal Voltage
1200
1100
Regulation Curve
Efficiency - Per Cent
100
80
60
40
20
1000
2000
3000
4000
5000
Output - Kilowatts
Curves Showing Efficiency and Regulation of 5,000 K.W. Acyclic Generator

V MECHANICAL DESIGN OF STEAM TURBINE

The steam turbine should be of slightly greater rating than the generator and will thus be calculated on the basis of 6,000 K. W. as the normal full load rating. The operating conditions, conforming with modern practice in well designed turbine plants will be assumed as follows:

Steam pressure - 190 pounds per square inch gage pressure or 205 pounds absolute.

Superheat - 200 degrees Fahrenheit.

Vacuum - 28.5 inches of mercury or .75 pounds absoluts pressure.

The turbine will be of the Curtis type having four pressure stages, with two velocity stages per pressure stage.

As indicated in the general method, the values of heat, quality, and entropy may best be obtained from the chart or table and in the present design will be as follows:

Initial heat H_1 = 1310 B. t. u. per pound.

Initial entropy S_1 = 1.66 entropy units above 32°F

Final value of heat with adiabatic expansion

$$H_2 = 915 \text{ B. t. u.}$$

Theoretically, available heat drop with adiabatic expansion

$$H_1 - H_2 = 395 \text{ B. t. u. per pound of steam.}$$

A suitable value of efficiency in this case is say 66.5 percent, making a final value of available energy, .665 x 395 = 266 B. t. u. per pound. The water rate would be $\frac{3410}{266}$ or 12.8 pounds per K. W. hour.

The entropy with this friction loss will be increased (Figure 16) due to the total friction loss of 395 - 266 = 129 B. t. u., from the initial value of 1.66 to a final value of 1.86. This value is determined by adding the total loss - 129 B. t. u. - to the value of H_2 = 915 B. t. u., which is then H_2^1 = 1044 B. t. u. At this value of heat contents, and with the same final pressure of .75 pounds per square inch, the entropy is found to be approximately 1.86. Figure 16 is copied from the steam heat entropy chart to indicate the change in entropy and quality of steam in passing through the various stages. Assuming equal efficiencies in all stages, the drop per stage would be one-fourth of the total available heat or 66.5 B. t. u. per pound.

This heat drop, in a correctly designed nozzle will give the steam a "useful" velocity of 1820 feet per second. This velocity, being calculated from the available energy is an equivalent value which includes all blade and frictional losses and is thus lower than would actually exist.

In designing the blades, it is most important that the angles be accurately determined and thus in drawing the various velocity triangles, (Figure 15), an allowance for loss of velocity due to blade friction was made. Since the effect of friction is to decrease the velocities obtained in the guide wheel and blades, it was necessary to increase or correct this value of initial velocity. The increase is necessary. since the efficiency constant, upon which the velocity was based, made allowances for all frictional losses, and in obtaining the blade angles it is advisable to make as close an

approximation as possible to the actual conditions, since a slight error of blade angle would cause serious impact and eddy losses in the steam flow, which in turn would result in greatly increased losses and a correspondingly poor water rate.

The average velocity loss in the guides and wheels is usually 22% for 180 degrees of angle. With 20 degree inlet and 36 degree outlet angles as maximum values, the friction on this basis would be

$$22\,\frac{180 - (20 + 36)}{180} = 15 \text{ percent approximately.}$$

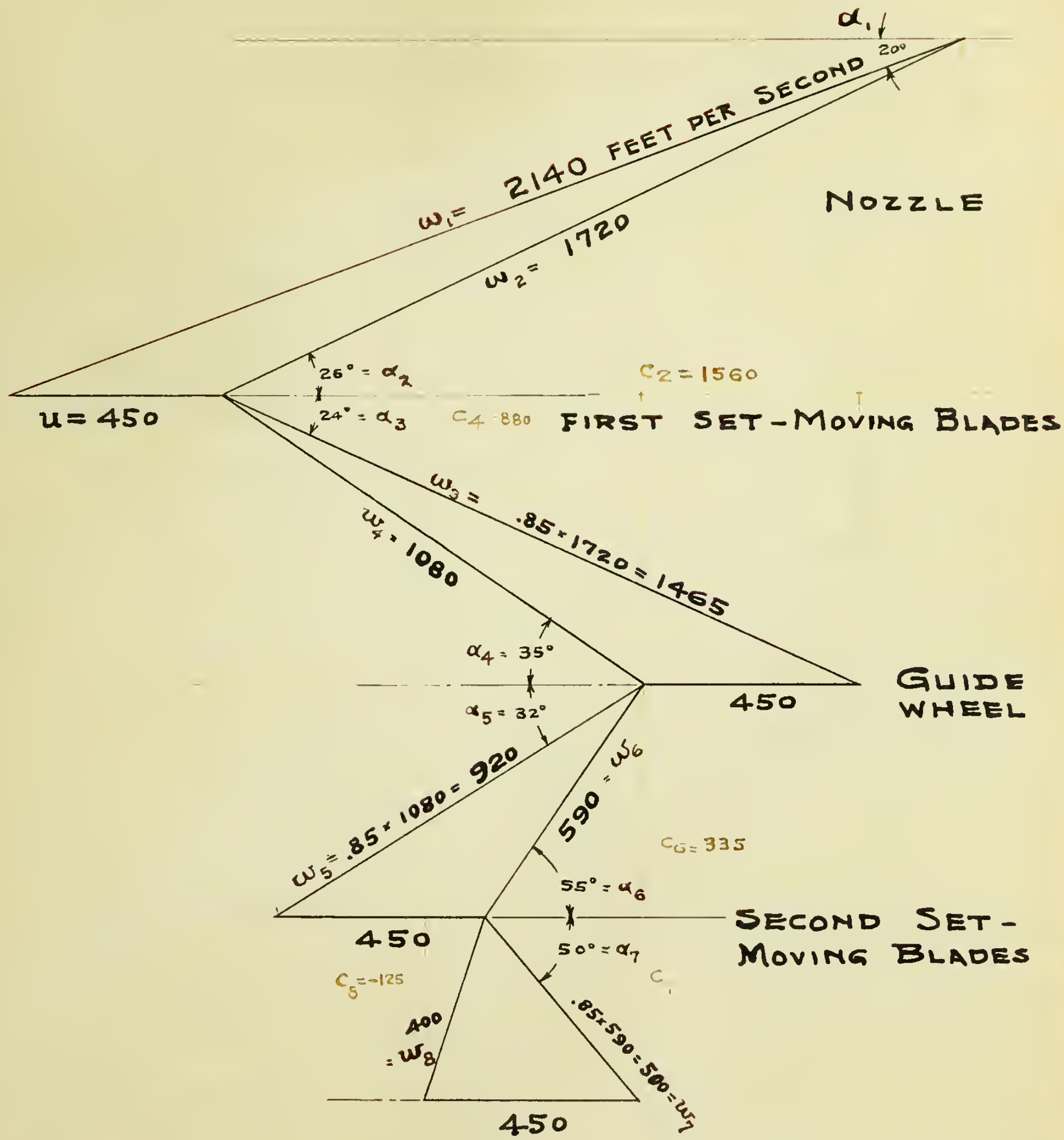
The other losses, making up a total of 34 percent, as assumed, might be distributed as follows:

Exhaust velocities	8	percent.
Churning and bearing friction	6	"
Blade friction	15	"
Leakage, etc.	5	"
Total	34	"

The initial velocity corrected for blade friction will then be $\frac{1.00}{.85}$ x 1820 or 2140 feet per second, and each blade velocity will thus be reduced by the factor .85, calculated above in the form of 15 percent loss.

Standard practice is to use an angle of 20 degrees for the nozzles and to make the inlet and outlet angles approximately equal for the other blades, Figure 15.

It is evident from the velocity diagram that a major part of the steam energy is extracted by the first wheel, since the change of velocity is a maximum at this point. The exit velocity ω_8 is made as small as possible and is allowed to



α_1
$20°$
$\omega_1 =$ 2140 FEET PER SECOND
NOZZLE
$\omega_2 =$ 1720
$26° = \alpha_2$
$C_2 = 1560$
$u = 450$
$24° = \alpha_3$
C_4 880
FIRST SET-MOVING BLADES
$\omega_3 =$ .85 × 1720 = 1465
$\omega_4 =$ 1080
$\alpha_4 = 35°$
$\alpha_5 = 32°$
450
GUIDE WHEEL
$\omega_5 =$ .85 × 1080 = 920
590 = ω_6
$C_6 = 335$
$55° = \alpha_6$
450
SECOND SET-MOVING BLADES
$50° = \alpha_7$
$C_5 = -125$
400 = ω_8
.85 × 590 = 500 = ω_7
450

FIG. 15.

have a slight forward component of velocity which indirectly reduces rotational losses and aids the steam in entering the next set of nozzles.

In the Curtis turbine, the steam expansion is practically confined to the nozzles and thus, since a constant efficiency was assumed throughout the turbine, the nozzle velocities of the various pressure stages will be equal and the only difference between the guide wheels and blades will be a change in the area of the steam channel to correspond with the decreasing pressure and increasing volume.

The specific volume of the steam in any given pressure stage is practically constant so that the channel area must increase in the same ratio that the velocity decreases. This fact explains the increase of channel area in a pressure stage and provides a means of calculating the channel area at any point after the nozzle has been designed and the steam area of the first row of blades determined.

The pressures are found by dividing the theoretical drop into four equal parts and finding the corresponding pressures from the chart. In this case they are: 205 pounds initial, 75 pounds first stage, 22 pounds second stage, 4.6 pounds third stage, and .75 pounds (per square inch absolute) fourth stage.

By connecting the points of initial and final condition of the steam on the chart (Figure 16), it is possible to determine the values of quality and entropy for the intermediate pressure stages and thus, with the pressures determined the specific volumes of the steam may be found and the nozzles

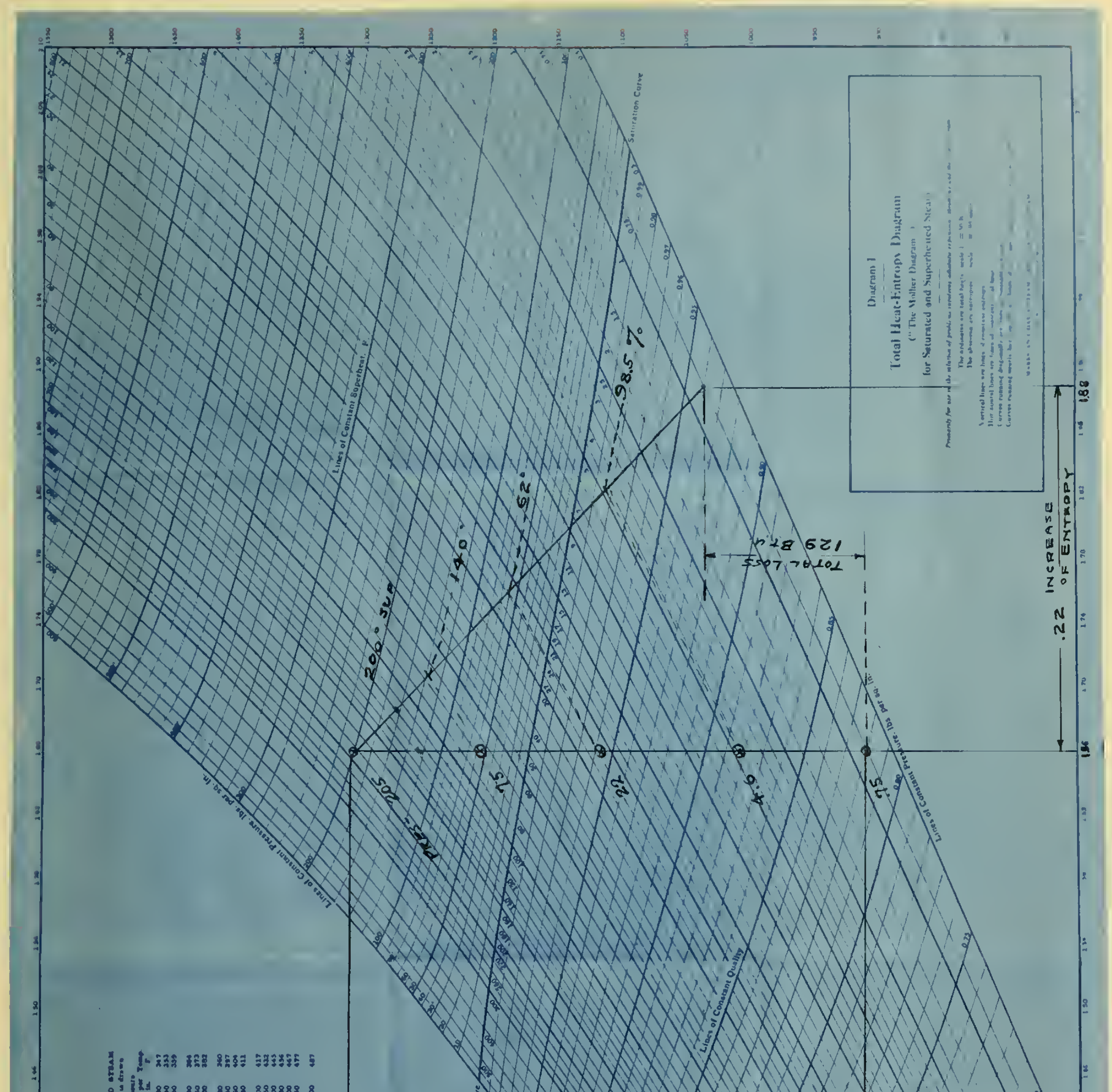

Total Heat-Entropy Diagram
for Saturated and Superheated Steam
Lines of Constant Superheat
Lines of Constant Pressure
Lines of Constant Quality
Saturation Curve
TOTAL HEAT DROP
266 B.t.u. per lb
TOTAL LOSS
129 B.t.u
.22 INCREASE OF ENTROPY
200° SUP
98.5 %

FIG. 16

designed for the various stages. By this method, a constant increase of entropy is assumed which, although not strictly accurate, is sufficient for an approximation.

A summary of the steam conditions in the various stages might be made as follows:

Initial - Pressure - 205 pounds per square inch absolute.
Quality - 200 degrees absolute (superheat)
Specific volume - 2.97 cubic feet per pound.

First Stage - Pressure - 75 pounds absolute.
Quality - 140 degrees superheat.
Specific volume - 7.09 cubic feet per pound.

Second Stage - Pressure - 22 pounds absolute.
Quality 62 degrees superheat.
Specific volume - 20.2 cubic feet per pound.

Third Stage - Pressure - 4.6 pounds absolute.
Quality - 98.5 percent.
Specific volume - 79 cubic feet per pound.

Fourth Stage - Pressure - .75 pounds absolute.
Quality - 94 percent.
Specific volume 410 cubic feet per pound.

Having determined the steam conditions throughout the stages, the next consideration is the design of nozzles, which depends upon the total flow of steam, its specific volume and the drop of pressure from stage to stage.

The total steam flow may be found from the calculated steam consumption and the full load output. In this design, the flow will be 6000 x 12.8 = 76,900 pounds per hour, or 21.4 pounds per second.

Assuming a total of twenty high-pressure nozzles, each at full load will receive 1.07 pounds of steam per second at a pressure of 205 pounds per square inch and a specific volume of 2.97 cubic feet per pound. In general, the nozzle area at any point will depend upon the fact that the volume passing any section is equal to the product of the velocity and the sectional area at that point.

Let G = number of pounds of steam flowing through each nozzle per second.

v = volume of steam - cubic feet per pound

ω = velocity in feet per second

F = section al area - square feet

F_1 = sectional area - square inches.

In the present case the quantities G, v and c, are known for any nozzle.

Then

$$F\omega = Gv$$

and

$$F = \frac{Gv}{\omega} \text{ when } F \text{ is in square feet,}$$

or

$$F_1 = \frac{144\ G\ v}{\omega} \text{ when } F_1 \text{ is in square inches.}$$

In this stage, G = 1.07 pounds per second. The volume changes with pressure, and since the pressure in the throat of the nozzle is .58 of the initial value (or 119 pounds) the conditions of volume and velocity may be found from the tables or heat chart, following the assumed path of entropy change as in Figure 16. Thus at 119 pounds per square inch, the superheat is 165 degrees, while the heat drop (actual) is 58 B. t. u. (since the loss is negligible at entrance), giving

a value of $\omega = 223.7 \sqrt{58}$

$= 1700$ feet per second in nozzle throat.

Also, at 119 pounds and 165 degrees superheat, the specific volume is $v = 4.74$ cubic feet per pound.

The first stage nozzle throat area may now be determined as

$$F_1 = \frac{144 \times 1.07 \times 4.74}{1700} \text{ square inches}$$

$$= .43 \text{ square inches.}$$

Assuming a round section, the diameter will be .74 inch.

The nozzle outlet area is determined in a similar manner, the value of G remaining constant. In this case, the steam conditions are those of the first stage, or

$v = 7.09$ cubic feet per second.

$\omega = 1820$ feet per second (available)

The outlet area is then

$$F_1 = \frac{144 \times 1.07 \times 7.09}{1820} = .60 \text{ square inches}$$

and if round, would be .875 inches in diameter. With 10 degrees nozzle divergence, the nozzle length will be

$$l = \frac{.875 - .74}{2 \tan 5^\circ} = \text{about } 3/4 \text{ inch}$$

The actual length would be longer, since the nozzle has a diagonal outlet at an angle of 20 degrees, but the dimension calculated would be the shortest length on the wheel side, while that of the opposite side would be 3-1/4 inches, approximately.

The first set of nozzles would thus occupy only a

small part of the wheel periphery, for if placed three inches on centers they would occupy a total of five feet or about one-sixth of the total periphery. However, the specific volume of the steam increases rapidly and in the last nozzles would require full peripheral admission with a very wide channel to accommodate the large steam volume.

In the first stage, as calculated, the area of the steam channel of the first wheel will be determined by total steam jet area in the five feet covered by the nozzles. Thus in a peripheral length of five feet, the first wheel must have an open channel area of at least 20 x .6 or 12 square inches. The blade lengths may be then approximated as follows:

Assuming that .7 of the periphery is effective steam channel, then the blade length will be

$$l_1 = \frac{12}{.7 \times 60} = \text{approximately } .3 \text{ inch.}$$

The second wheel of the first stage will have a longer blade to provide the increased area due to decreased steam velocity, the specific volume remaining practically constant. Thus with a decrease of velocity from 2000 to 500 feet per second, the area must be increased to four times its original value. The steam, however, is now probably distributed over more of the periphery so that the increase of blade length will not be four times, but will rather follow a gradually diverging path similar to a nozzle section, the object being to keep the steam as much aspossible in the blade channels and prevent the excessive windage losses caused by side leakage of steam.

The nozzles of the remaining stages are calculated in the same manner and the steam admission, as indicated, will occupy a larger part of the moving buckets. The minor details of blade thickness, width and material, although beyond the scope of this general design, are important and practically determine whether the turbine will show the assumed efficiency when tested. The object in arranging the blades is to obtain the greatest effect with the least friction. This is evidently necessary, since too small a blade causes eddy losses while too large a blade causes friction losses. The mean value is a matter of experience.

An approximate check may be made in this design by calculating the work of the first stage and comparing it with the available energy assumed.

The work extracted from the steam jet by the buckets may be calculated if the change of velocity in the direction of motion is known. Thus, with a steam flow of 1 pound per second and a total change of velocity, as indicated, of c feet per second, the force exerted, being equal to product of mass and acceleration, would be

$$F^{1} = \frac{c}{g} \quad \text{where } g = \text{acceleration of gravity.}$$

The mass in this case is $\frac{1}{g}$ and the change of velocity per second is $c = c_1 - c_2$ or the difference between initial and final velocities if they are in the same direction or their sum if in opposite directions.

Thus in the present design, the axial components of the various velocities corresponding to ω_1, ω_2, etc. given are

$c_2 = 1560$ $\qquad$ $c_6 = 335$

$c_4 = 880$ $\qquad$ $c_8 = -125$

Thus $\quad F^1 = \dfrac{1560 + 880 + 335 - 125}{32.2}$ per pound of steam.

$$\text{Work} = F^1 u = \frac{450 \Sigma c}{g}$$

$$\text{and B. t. u.} = \frac{\text{Work}}{778} = \frac{450 \Sigma c}{778 \text{ g.}}$$

Then the available energy per pound of steam is

$$\text{B. t. u.} = \frac{450 \times 2550}{778 \times 32.2} = 45.8$$

The calculated was 66.5 B. t. u. or $\frac{45.8}{66.5} = 69$ percent of the theoretical. A slight mechanical loss would decrease this to 66 percent, indicating that if the velocity diagram indicates ideal oonditions, the efficiency as calculated above bears out the assumption of mechanical efficiency.

The calculations of the shaft and wheels will be taken up as "Practical Problems" in the following pages.

VI ARRANGEMENT AND OTHER PRACTICAL PROBLEMS

The advantages of the vertical shaft in this type of unit have been outlined only in their relation to the mechanical design. Another most important advantage, especially from the standpoint of the owner and operator, is the unusually small floor space occupied by this type of unit as compared with a reciprocating engine or even a horizontal turbine. Thus, the values of floor space in square feet per kilowatt output for modern power plants varies from approximately one square foot with vertical turbines, to three square feet or more with horizontal reciprocating engines, indicating the great advantage of this detail of arrangement from the standpoint of the cost of land. Another factor in the space-efficiency of this unit is the use of the turbine base as a condenser, which thus materially reduces the floor space covered by the so-called auxiliaries.

Among the many problems encountered in steam turbine design, the one which above all others must be carefully considered, is that of ample strength in the rotor at the excessive peripheral speeds customary. The working stresses thus produced are far in excess of the maximum allowable stresses encountered in ordinary engineering practice, and thus necessitate the use of special engineering materials such as alloy steels having unusual tensile strength combined with toughness and sufficient ductility.

An interesting fact in connection with rotating

elements, and especially in the case of a hollow drum, is that the tangential stress in the rim section depends only upon the value of peripheral speed. Referring to Figure 17, page 62, which shows a semi-circular section of a drum rim with centrifugal and tangential forces acting upon it; the section for convenience is assumed to be of unit length axially, and to be in a condition of equilibrium under the influence of the various forces acting upon it. The tangential forces T + T balance the resultant of all centrifugal forces OF acting on the half ring section.

$$\text{Then} \quad 2\,T = \int_0^{\pi} d\,F \sin\theta\, d\,\theta.$$

Then

dF, the element of centrifugal force may be expressed in terms of the angle θ and K = specific mass (or $\frac{\text{density}}{g}$) as follows:

$$dF = (K\,r\,d\,\theta\,t)\,r\,\omega^2$$ where ω is the angular velocity in radians per second.

Substituting,

$$2\,T = K\,r^2\omega^2\,t \int_0^{\pi} \sin\theta\,d\,\theta$$

or $T = K\,t\,v^2$ where v = peripheral velocity in feet per second.

The stress $S = \frac{T}{t}$ with unit length of section area,

and $S = K\,v^2$.

If v is in feet per second, K is $\frac{\text{wt. per cu. ft.}}{g}$. Then for steel, the stress in pounds per square inch

$$S = \frac{490}{32.2 \times 144}\,v^2 = .109\,v^2.$$

A graph of this equation is shown on page 64, which also indicates that a peripheral speed of 400 feet per second is

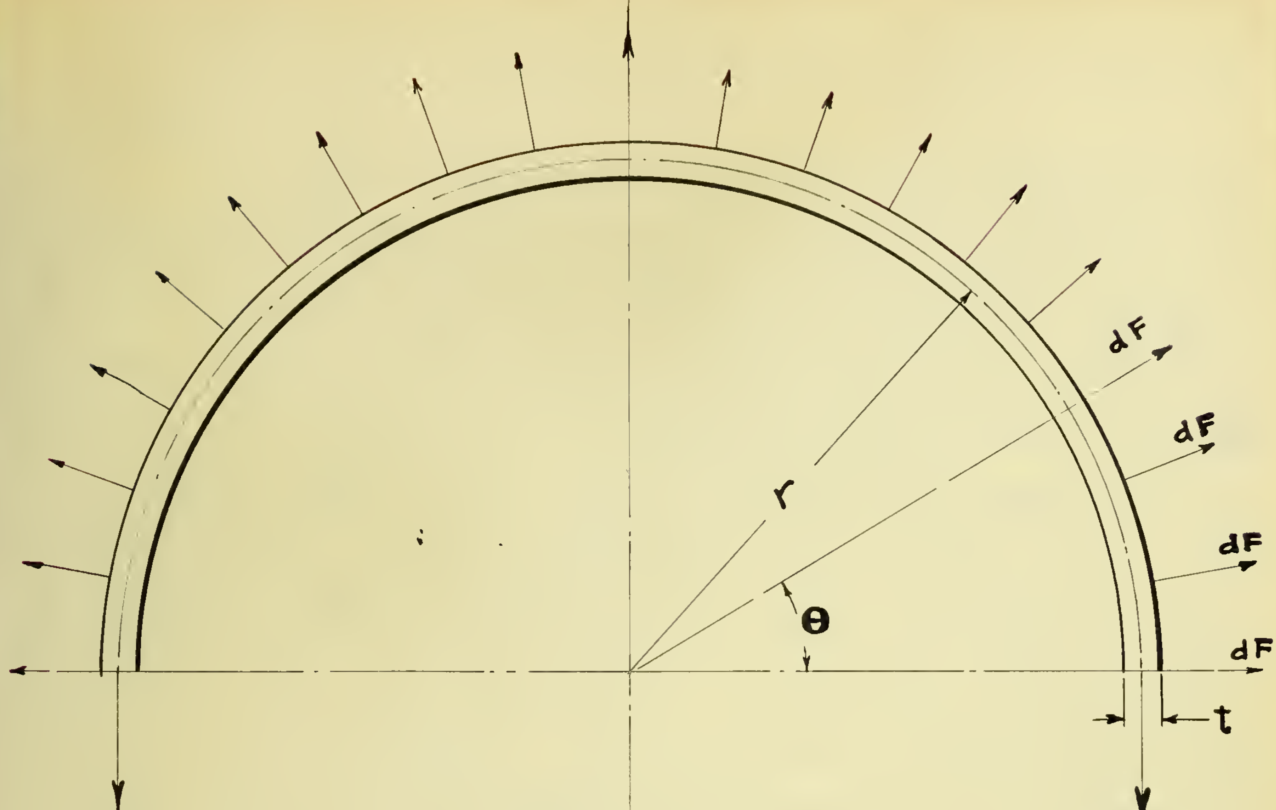


FIG. 17

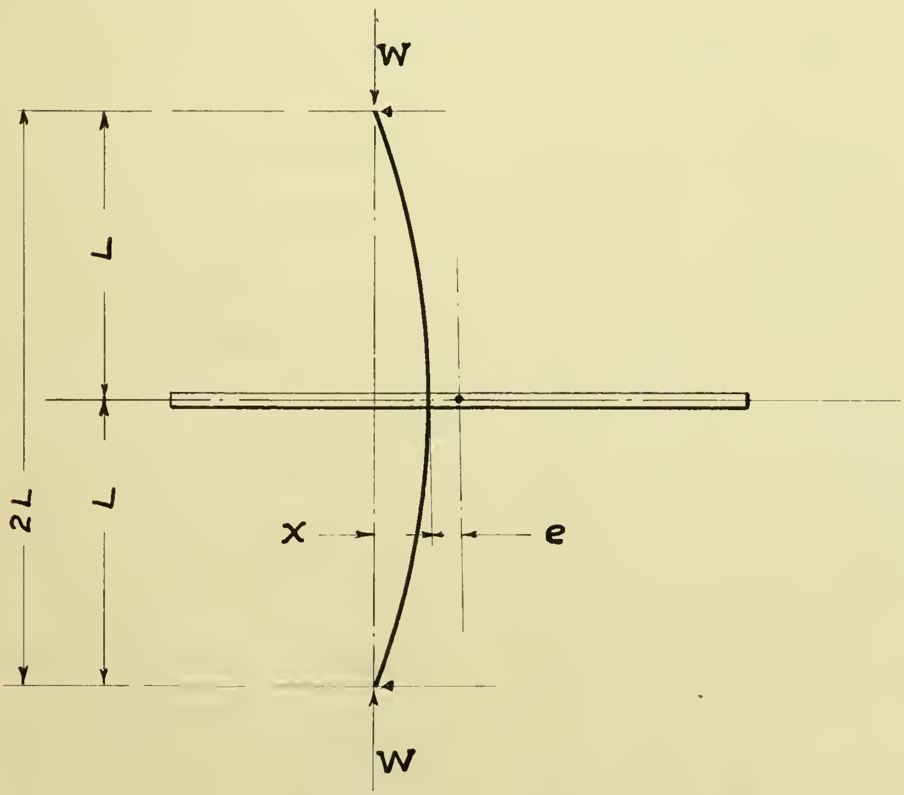


FIG. 18

approximately the maximum allowable speed with ordinary good steel.

Next in importance to the question of obtaining sufficient strength in the rotor is the problem of avoiding what is known as the "critical frequency" of the shaft. This frequency occurs at such a speed that the centrifugal force balances the resistance of the shaft to bending, and thus a free vibration takes place. As it is practically impossible to build turbine rotors so accurately that the mass center of the wheels coincides with the geometric center of the shaft and there is thus an eccentricity e - Figure 18,- between mass center and shaft axis. The conditions of the present design are approximated by assuming an eccentricity e, a deflection x due to the centrifugal force of the disc, and a force W corresponding to the weight of the acyclic rotor above the turbine shaft.

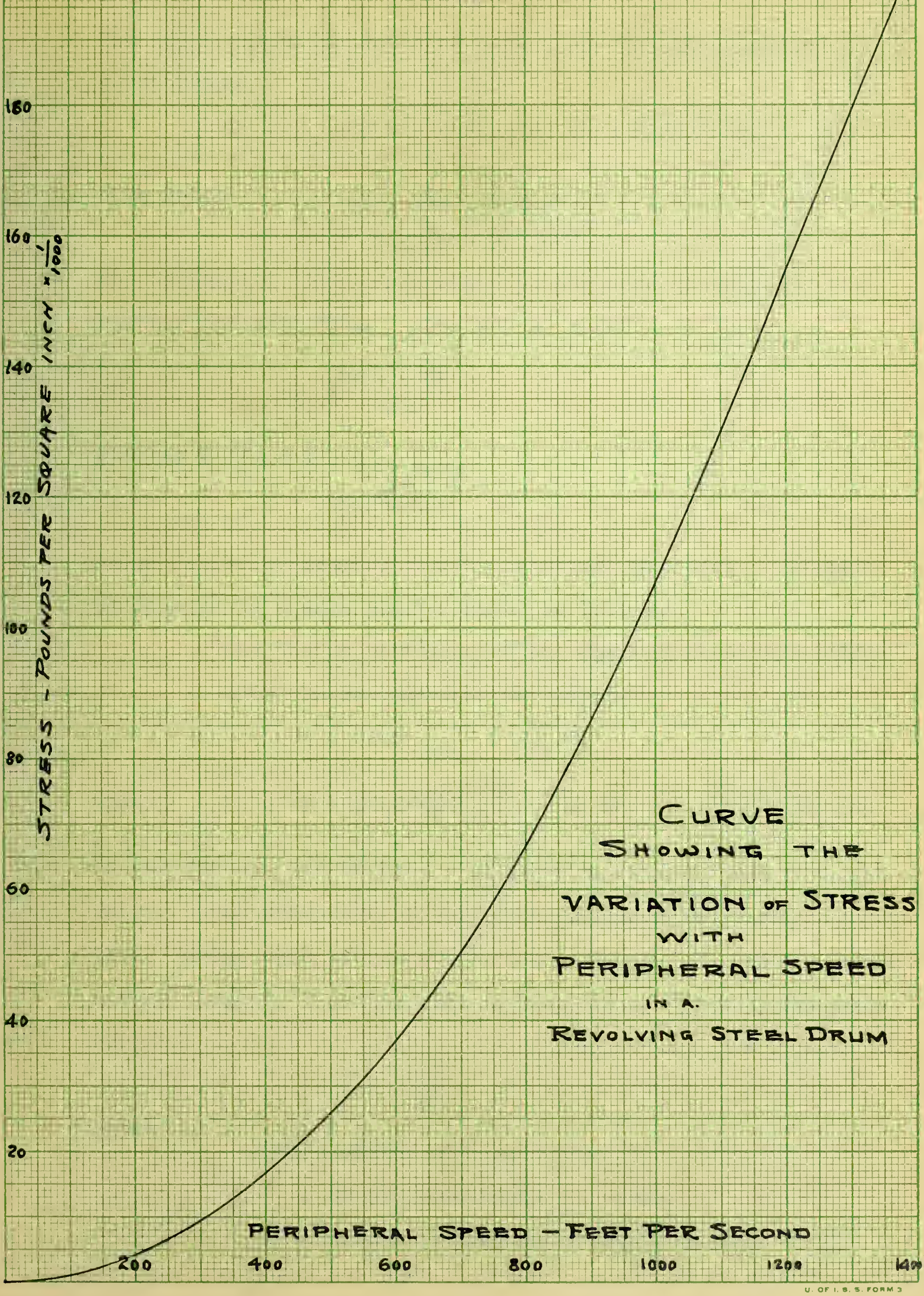
Then, for a condition of equilibrium, the total resisting force may be assumed proportional to the deflection x. Thus if α is a constant depending upon shaft conditions, bearings and other constants, the resisting force

$$P = \alpha x.$$

The force causing deflection is then equal and opposite to P and is made up of two components, - the centrifugal force P_1 of the disc with mass center at radius $(x + e)$ and an equivalent force P_2, acting in a direction co-linear with P_1, to balance the effect of the end load W.

Thus $$P_2 = \frac{2\,W\,(x + e)}{L}$$ where 2 L is the total shaft length.

But $P_1 = m\,(x + e)\,\omega^2$


Curve
Showing the
Variation of Stress
with
Peripheral Speed
in a
Revolving Steel Drum
Stress - Pounds per Square Inch × 1/1000
Peripheral Speed - Feet per Second
20
40
60
80
100
120
140
160
180
200
400
600
800
1000
1200
1400

The total resisting force is then from the conditions of equilibrium,

$$P_1 + P_2 = \alpha x = m(x + e)\,\omega^2 + \frac{2W(x + e)}{L}$$

from which

$$x = \frac{m L \omega^2 e + 2 W e}{L - (m L \omega^2 + 2W)}$$

The deflection thus increases with the angular velocity and becomes infinite when $L = m L \omega^2 + 2W$

or
$$\omega = \omega_K = \sqrt{\frac{\alpha L - 2W}{m L}}$$

Putting $2\frac{W}{L} = C$

$$\omega_K = \sqrt{\frac{\alpha - C}{m}}$$

which is the expression for the critical velocity. It may be reduced to a more useful basis as follows:

Let $G = mg$ = weight of the wheel or disc in pounds.

Also the revolutions per minute

$$n = 60 \times \frac{\omega}{2\pi} = \frac{30\,\omega}{\pi}$$

Then
$$n = 54.2\sqrt{\frac{\alpha - C}{G}}$$
as the critical velocity in revolutions per minute. It remains only to determine the values of α which are derived directly from the theory of beams.

Thus, in the case of a shaft having a length 2L with a center lateral load of P pounds, the deflection is given by the relation

$$x = \frac{PL^3}{6\,EI}$$

where

E = modulus of elasticity

= 30,000,000 nearly for steel.

I = moment of inertia of shaft section = $\frac{\pi d^4}{64}$

if d = shaft diameter; or for a shaft fixed at the ends

$$x = \frac{PL^3}{24\ E\ I}$$

Then assuming the latter case,

$$= \frac{P}{x} = \frac{24\ EI}{L^3} \text{ for this condition.}$$

The quantity C may be approximated in the equation,

$$C = 2\ \frac{W}{L} \text{ by assuming from the general}$$

design that W = 120,000 pounds

$$L = 7 \text{ feet} = 84 \text{ inches.}$$

$$C = \frac{2\ \omega}{L} = 2,860.$$

Evaluating and substituting in the equation

$$n = 54.2 \quad \frac{-\ C}{G} = \text{critical R. p. m.}$$

$$= \frac{24 \times 30,000,000 \times \pi d^4}{64 \times 638,000} = 55.5\ d^4$$

The wheel weights of 8 wheels 10.4 feet in diameter and 2.6 inch mean thickness.

$$G = 73,000 \text{ pounds.}$$

Then

$$n = 54.2 \quad \frac{55.5\ d^4 - 2,860}{73,000}$$

or

$$n^2 = 2.22\ d^4 - 116$$

A graphical solution of this equation is shown on page 68. It is evident that the constant term has little effect with large shaft sizes and thus in this condition

$$\frac{n}{d^2} = 1.49 \text{ for large diameter.}$$

The assumption of a concentrated wheel of equivalent weight

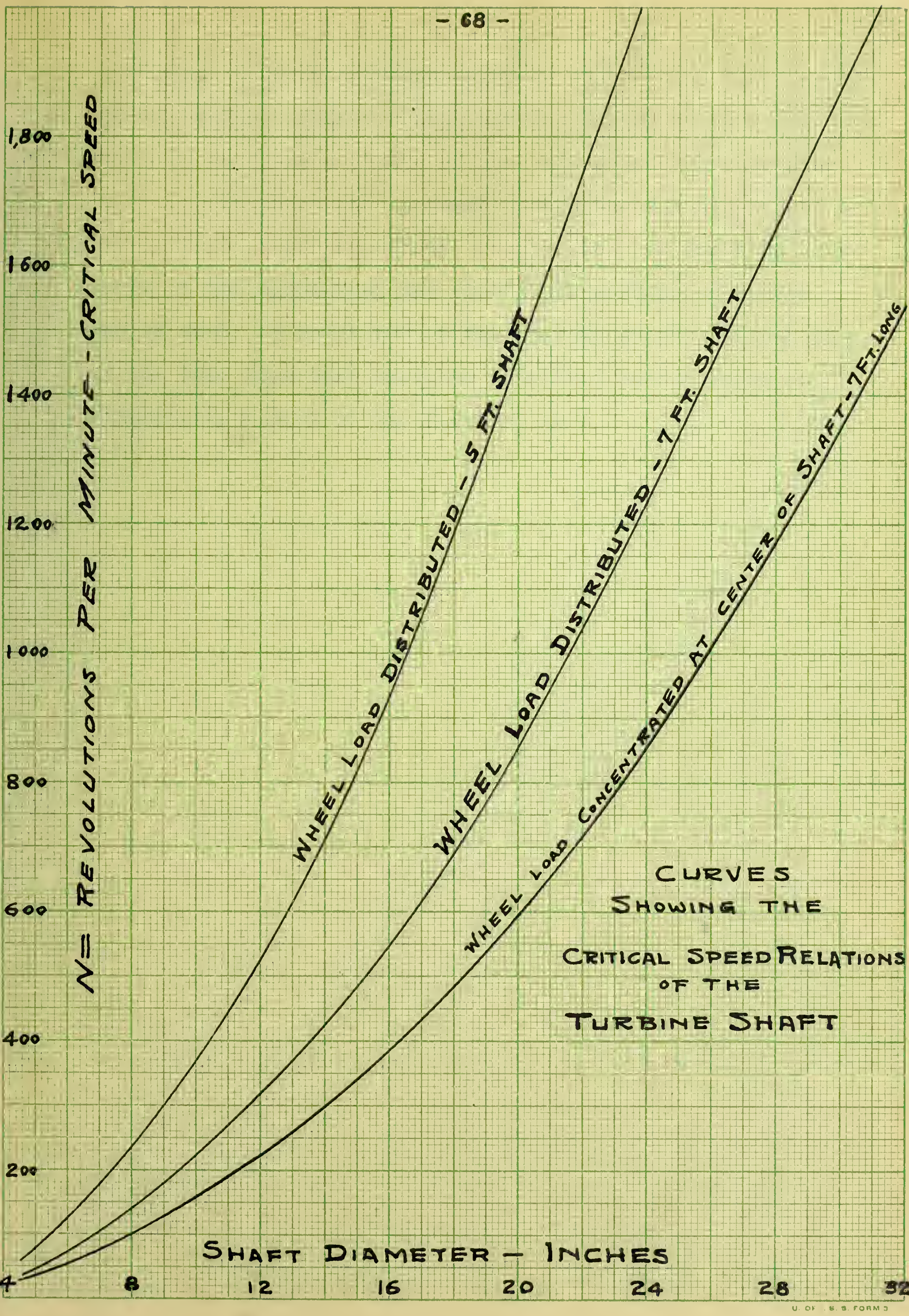
placed in the center of the shaft gives values of shaft diameter which are evidently larger than necessary, since in the actual turbine the wheels are distributed. With a distributed load, the deflection is reduced to one-half of its former value and the other curve shows the effect upon the critical frequency of decreasing the center load.

The critical frequency chosen should be in excess of a value which would be attained if the turbine should "run away". On account of wheel friction, which varies as the third power of the speed, it is probable that under these, no load or "run away", conditions the maximum speed would not exceed twice normal speed. This value, in fact, could scarcely be reached, since the normal blade friction is about 15 percent. The maximum speed would then be

$$N_m = \sqrt[3]{\frac{100}{15}} = 1.88 \text{ times normal full speed.}$$

With a value of 1500 R. p. m. as the critical speed, the shaft diameter would be approximately 25 inches, according to the assumptions made. However, the shaft could not be definitely decided until the entire blade system had been designed. The actual loading would then be known and a more accurage determination could be made. Thus with a shaft five feet long the diameter for this critical velocity would be only 20 inches, showing the effect of shortening the shaft, and the necessity of more accurate assumptions.

There are many other practical problems encountered in the design and manufacture of a unit of the present type, but the scope of this thesis will not permit of their introduction here. The general problem of speed control, or


N = Revolutions per Minute - Critical Speed
1,800
1600
1400
1200
1000
800
600
400
200
Shaft Diameter — Inches
4
8
12
16
20
24
28
32
Wheel Load Distributed - 5 Ft. Shaft
Wheel Load Distributed - 7 Ft. Shaft
Wheel Load Concentrated at Center of Shaft - 7 Ft. Long
Curves
Showing the
Critical Speed Relations
of the
Turbine Shaft
U. OF I. S. S. FORM 3

governing of steam turbines would in itself be a large undertaking, while a corresponding problem on the electrical side would be a complete design of the brushes and attachments.

In short, the present thesis had for its object a study of the inter-relations between the more important features of the mechanical and electrical methods of design, and as such, it could not in the present scope, include many of the less important features of either design which are essentially a part of the finished unit.